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SOLAR HEATING OF BUILDINGS AND DOMESTIC HOT WATER. REVISION.(U)
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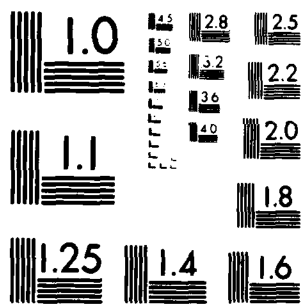
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Solar Heating of Buildings and Domestic Hot Water

Technical Report 877

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FOREWORD

Conversion to solar heating at Naval facilities, where cost effective, could have a significant impact on dollar and fuel savings. The amount of available sun, the cost of equipment, the cost of available fossil fuels, and building heating loads must be considered by facility engineers in designing cost-effective solar heating systems. This document presents guidelines for engineers to make preliminary design and cost analyses and to prepare specifications for bidders on solar systems.

We are working with NAVFAC towards the development of a design manual for utilization of solar energy. This document, a revision of CEL TR 835, represents a long step in that direction. An important feature of this report is the use of worksheets for stepwise calculation of total installation costs, solar collector parameters, domestic hot water demand, monthly solar radiation yields, cost avoidance for fossil fuel, and other costs associated with solar heating installations. This revision has incorporated some additional information on cooling with solar energy and passive solar construction.

Recommendations or modifications to this manual based on experience in using it should be submitted to: Code L03AE, Civil Engineering Laboratory, Naval Construction Battalion Center, Port Hueneme, CA 93043.

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1.0 INTRODUCTION

1.1 Scope

This report presents design criteria and cost analysis methods for the sizing and justification of solar heat collectors for potable water heaters and space heaters. Sufficient information is presented to enable engineers to design solar space and water heating systems or conduct basic feasibility studies preparatory to design of large installations. Both retrofit and new installations are considered. This report has been substantially revised from the previous edition (Beck and Field, 1977). However, most of the revision is in Section 2.0, where more material of an exploratory nature has been added. Section 3.0, which contains the calculation method and worksheets, is largely the same, except that the economic analysis has been revised and new tables have been added to provide a self-contained source of meteorological data and collector test data. Previous users will have no trouble using the worksheets of Section 3.0.

If information is desired on "how" to select, size, or orient a solar system, this information will be found in Section 2.0. It is intended that this section be used as a reference in which the user may find information on many topics relating to solar energy systems. New sections include solar cooling, passive systems, collector fluids, and heat pumps. Other sections have been revised to include more information.

1.2 Related Criteria

Certain criteria relating to space heating and domestic hot water (DHW) heating systems appear elsewhere and are listed below.

a. The Department of Defense general requirements are found in the Construction Criteria Manual, DOD 4270.1-M.

b. Some portions of Design Manual DM-3 relating to heating and hot-water systems pertain to this manual. These and other relevant sources of applicable criteria are listed below:

Subject	Source
Plumbing Systems	Chapter 1
Heating Systems	Chapter 3
Architectural Criteria	Chapter 5
Electrical Criteria	Chapter 5
Hazards & Safety Precautions	Chapter 5
Insulation	Chapter 5
Structural Criteria	Chapter 5
Central Heating Plant	Chapter 8
Corrosion Protection	Chapter 9
Water Conditioning	Chapter 9
Housing & Building Designs (definitive)	NAVFAC P-272
Weather Data	NAVFAC P-89

c. Standards and performance criteria relating to solar heating systems have been evolved by government agencies and various associations and institutes. The most widely used are listed below:

Subject	Document
Solar Collector Instantaneous Performance	ASHRAE Standard 93-77, "Methods of Testing to Determine the Thermal Performance of Solar Collectors"
Thermal Storage Devices	ASHRAE Standard 94-77, "Methods of Testing Thermal Storage Devices Based on Thermal Performance"
Complete Solar Collector Performance Evaluation	National Bureau of Standards, NBSIR 78-1305A, "Provisional Flat Plate Solar Collector Testing Procedures: First Revision"
Testing Solar Hot Water Heaters	ASHRAE Standard 95, "Methods of Testing Solar Energy Potable Water Heaters"

Subject	Document
Testing Swimming Pool Solar Collectors	ASHRAE Standard 96, "Methods of Testing to Determine the Thermal Performance of Liquid Solar Collectors to Heating of Swimming Pools"
Solar System Performance	National Bureau of Standards, NBSIR 76-1187, "Interim Performance Criteria for Solar Heating and Cooling Systems in Commercial Buildings"
Property Standards for Solar Systems	HUD Report 4930.2, "Intermediate Minimum Property Standards Supplement, Solar Heating and Domestic Hot Water Systems"
Property Standards Developed for HUD Domestic Hot Water Initiative	National Bureau of Standards, NBSIR 77-1272, "Intermediate Standards for Solar Domestic Hot Water Systems/HUD Initiative"
Solar Collector Certification and Labeling	Standard 910, "The Air Conditioning and Refrigeration Institute (ARI) Certification Program for Solar Collectors"
Solar Collector Certification and Labeling	Solar Energy Industries Association, Standard PCS 1-79, "S.E.I.A. Certified Thermal Performance Rating Standard for Solar Collectors"
Building Code (Second Draft, September 1979)	Council of American Building Officials, DOE/CS/4281-1, "Model Document for Code Officials of Solar Heating and Cooling of Buildings"
Product Safety Standard (not completed) (See HUD report 4930.2 for current safety standards.)	National Bureau of Standards, NBSIR 78-1143A, "Plan for the Development and Implementation of Standards for Solar Heating and Cooling Applications"

In addition to these standards, there are installation standards published by the Sheet Metal Air Conditioning Contractors' National Association (SMACNA), plumbing standards published by The International Association of Mechanical and Plumbing Officials (IAMPO), and various state building codes.

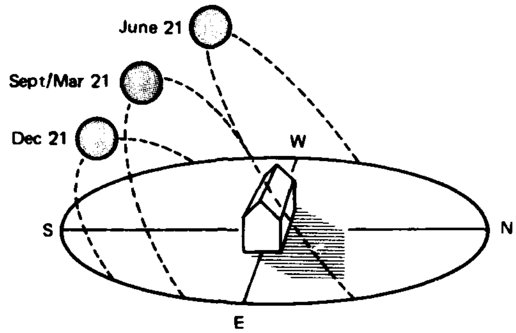


Figure 1-1. The sun's path across the sky at specific times of the year.

1.3 Solar Energy

1.3.1 Solar Radiation. Energy from the sun is received by the earth as electromagnetic radiation. Most of the energy is received in the visible and infrared portions and a small amount as ultraviolt radiation. North of the Tropic of Cancer (23°N latitude), the sun makes a daily arc across the southern sky from east to west as shown in Figure 1-1. For a typical location at 32°N latitude the sun would be 81.5° above the southern horizon or nearly overhead at noon (solar time) on June 21 while on December 21 it would be only 34.6° above the horizon (Barnaby et al., 1977).

Solar insolation (I) is measured in Langleys (L). One Langley equals 3.688 Btu/ft^2. The amount of solar energy that exists outside the atmosphere, often called the solar constant, is 116.4 L/hr or 429.2 Btu/ft^2-hr. At most 70% to 80% of this amount will strike the earth's surface, the remainder being absorbed or reflected in the atmosphere. Monthly average and yearly average daily insolation data for numerous Naval installations are given in Table 1-1. In general, the higher the latitude, the less insolation is received on a horizontal surface.

Table 1-1. Average Solar Radiation Intensities, Langleys/Day (Horizontal Surface)

Radiation Data From	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec	Year
Annette, AK *	63	113	231	360	457	466	481	352	266	122	59	40	251
Page, AZ *	294	367	516	618	695	707	680	596	516	402	310	243	495
Yuma, AZ	305	401	517	633	703	705	652	587	530	442	330	271	506
Davis, CA *	158	256	402	528	636	702	690	611	498	348	216	148	433
Fresno, CA *	186	296	438	545	637	697	668	606	503	375	241	160	446
Inyokern, CA *	312	419	578	701	789	836	784	738	648	484	366	295	579
Los Angeles, CA *	243	337	446	518	517	594	645	579	505	365	277	228	442
Pasadena, CA	251	333	439	509	569	580	634	599	482	366	271	236	439
Riverside, CA *	271	362	468	526	608	666	652	603	521	400	309	260	470
San Diego, CA	265	343	428	464	493	510	547	499	446	361	284	245	407
Washington, DC *	159	230	320	403	447	558	529	462	367	281	211	147	343
Gainesville, FL *	278	367	445	539	586	544	520	508	444	368	318	254	431
Jacksonville, FL	267	346	423	514	556	525	522	476	383	331	274	230	404
Key West, FL	327	410	490	572	579	543	534	501	445	394	332	292	452
Miami, FL *	343	416	491	544	552	531	537	508	447	389	354	319	453
Pensacola, FL	250	321	405	509	562	568	537	509	430	394	278	224	416
Tallahassee, FL *	274	311	423	483	548	476	544	537	424	353	364	260	416
Atlanta, GA *	228	284	377	484	535	554	538	502	412	350	265	201	394
Griffin, GA *	238	302	388	519	577	580	559	523	437	372	288	210	416
Pearl Harbor, HI *	355	404	438	536	577	562	610	575	536	466	393	349	483
Lemont, IL *	171	232	326	390	497	553	527	486	384	265	157	131	343
Indianapolis, IN *	147	214	312	393	491	547	542	486	405	293	176	130	345
Louisville, KY	164	231	325	420	515	560	550	498	408	303	190	150	360
Lake Charles, LA *	239	304	396	483	554	582	521	506	448	402	296	232	414
New Orleans, LA	237	296	393	479	539	549	502	491	418	389	269	220	399
Boston, MA *	139	198	293	364	472	499	496	425	341	238	145	119	311
Portland, ME *	157	237	359	406	513	541	561	482	383	273	157	138	351
Annapolis, MD	175	243	340	419	488	557	542	469	383	294	189	155	355
Silver Hill, MD *	182	244	340	438	513	555	516	459	397	295	202	163	359
St. Cloud, MN *	170	251	366	423	499	541	555	491	360	241	146	123	348
Cape Hatteras, NC *	244	317	432	571	635	645	629	557	472	361	284	216	447
Sea Brook, NJ *	157	227	318	403	478	522	518	457	385	285	192	139	340
Trenton, NJ	173	244	343	424	491	546	540	469	389	294	195	155	355
Ely, NV *	238	333	464	564	624	708	648	608	519	393	287	220	467
Reno, NV	234	324	449	592	664	714	707	646	532	395	277	209	479
New York, NY *	146	210	312	378	455	526	518	492	361	262	160	128	324
Oklahoma City, OK *	255	317	407	498	540	623	610	588	484	379	284	237	435
Philadelphia, PA	175	242	347	425	493	554	538	465	388	293	191	152	355
State College, PA *	139	202	297	373	467	544	528	454	361	275	155	120	33[illegible]
Newport, RI *	155	231	330	395	489	538	517	449	380	273	175	141	33[illegible]
Charleston, SC	250	308	393	517	553	556	523	495	417	349	281	228	406
Nashville, TN	163	240	329	450	517	567	553	494	428	327	217	161	370
Brownsville, TX *	287	336	402	458	556	604	619	555	465	406	284	253	435
Corpus Cristi, TX	262	330	413	474	561	604	629	558	470	408	285	240	436
Dallas, TX	231	307	394	454	521	595	588	538	458	363	261	221	411
El Paso, TX *	331	432	549	655	715	730	670	639	575	462	367	313	536
Norfolk, VA	208	270	372	477	540	572	550	481	398	310	223	184	382
Seattle, WA *	70	124	244	360	446	471	501	431	310	174	90	59	273
Albrook A. B. Panama *	392	476	525	499	404	336	370	372	448	338	380	420	426
Wake Island *	438	518	570	623	644	648	636	623	587	530	485	399	558
San Juan, P. R. *	429	489	581	607	555	612	643	574	542	495	428	428	532
Taipei, Taiwan	186	216	261	312	381	393	400	412	341	340	296	225	314

* From "World Distribution of Solar Energy," University of Wisconsin Engr. Expr. Sta. Rpt no. 21, by G.O.G. Lof, J. A. Duffie, and C. O. Smith, July 1966.

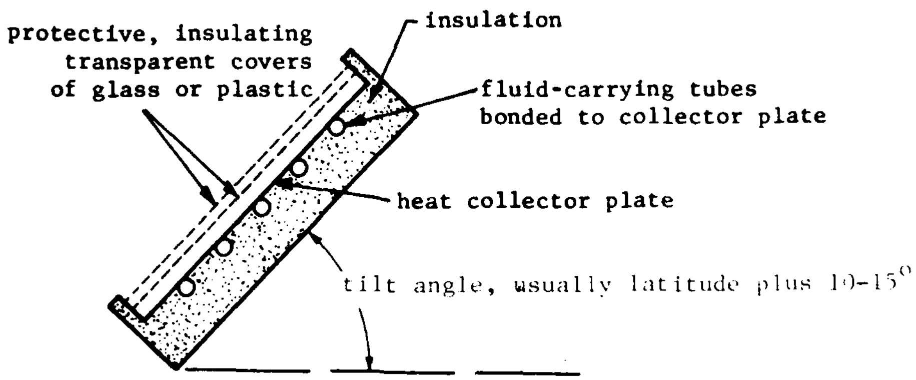


Figure 1-2. Schematic cross section of typical solar heat collector with heavy back insulation and two cover sheets.

1.3.2 Collecting Solar Energy. Collection of solar energy is based on the very high absorption of radiant energy by dull, black surfaces and on the "greenhouse effect." The latter refers to the ability of glass to transmit visible radiation but prevent the loss of heat from the collector plate which radiates at longer wavelengths (infrared frequencies). Glass (or plastic) cover plates are generally used over flat absorber plates to reduce heat loss (see Figure 1-2). The heated absorber plate may have a fluid (water, air or other) pass over it or through tubes attached to the plate. The fluid thus heated may be used to heat potable water, heat spaces or drive an absorption or Rankine cycle air conditioner.

The amount of solar energy collected by a solar collector depends on its efficiency, which is determined by how it is constructed, its configuration, and the choice of materials used. Standards are available as given in Section 1.2 which can test the instantaneous efficiency of a solar collector for a specified set of operating conditions.

Other parameters, not related to the physical characteristics of a solar collector, can affect performance. The atmosphere reduces the radiation received on the earth's surface and is also responsible for the

scattering of light which results in diffuse, as distinct from direct, solar radiation. The diffuse component may represent as much as 25%-30% of the total solar radiation depending on the weather conditions. Solar flat plate collectors absorb heat from the diffuse component as well as the direct. Thus, some heat is available on partly cloudy days. The reflectance of the ground (snow, sand, water, etc.) or nearby objects may also influence the amount of solar energy reaching a collector. Therefore, the amount of solar energy received at any location depends on the hour of the day, the day of the solar year, and the meteorological conditions. This amount can vary from about 50 Btu/ft^2-hr on a foggy winter day to as much as 300-375 Btu/ft^2-hr on a typical sunny summer day.

1.3.3 Solar Collector Orientation. Even though solar collectors can collect heat from the diffuse component of solar radiation, solar systems are designed to use the direct component. Direct radiation is in the form of parallel rays coming straight from the sun. To best capture this energy the solar collector should be tilted as shown in Figure 1-2 so that it is more nearly perpendicular to the solar rays. The "optimum" tilt angle varies even as the sun changes its position throughout the day and year. However, since the solar system cannot be continuously moved, some general rules can be stated:

1. For all year domestic hot water (DHW) heating use a tilt angle equal to the latitude.

2. For all year DHW heating and winter space heating use a tilt angle equal to the latitude plus 10-15 degrees.

3. For all year DHW heating, winter space heating, and summer cooling use same as no. 1.

4. For winter only space heating use a tilt angle equal to the latitude plus 10-15 degrees.

5. For summer only space cooling use a tilt angle equal to the latitude minus 10-15 degrees.

6. For summer only space cooling and all year domestic hot water heating use the same as no. 5.

In addition to choosing the best collector tilt angle, consideration must be given to the orientation of a collector (i.e., the direction the collector faces). Normally true south is the best and most frequent choice. However, slightly west of south (10 degrees) may be preferable in some locations if an early morning haze or fog is a regular occurrence.

Some deviations from these tilt and orientation angles are allowable without significantly affecting performance. As shown in Figures 1-3 and 1-4, the tilt angle may vary ±10 degrees and the orientation angle up to 20° either side of true south (National Solar Heating and Cooling Info Ctr, 1979). For these deviations the solar collectors would still collect 95%-100% of their rated capacity in most locations of the U.S. Additional deviations would require more collector area to capture the same amount of energy. As a very approximate rule of thumb, for each deviation of 10° beyond that shown in Figures 1-3 and 1-4 add 10% more collector area. If you must choose between an east roof and a west roof, use the west roof in the western coastal area. Other areas will require local weather considerations.

As important as collector location, is keeping the collectors out of the shade, especially between 9 a.m. and 3 p.m., when most of the useful energy collection occurs. In summary, although many buildings will not have a "perfect" solar orientation, there can still be many places with good solar energy potential.

1.3.4 Advantages and Disadvantages. Solar energy is inherently nonpolluting, provides substantial freedom from the effects of fuel price increases, and saves valuable fossil fuels. Disadvantages are that collectors perform poorly in cold cloudy weather, when most needed; and room heat exchangers and industrial unit heaters must be larger than in conventional systems due to the relatively low temperature of heating fluid. The disadvantages may be circumvented by good design;

where fuel costs are high enough (as discussed in the examples, Section 4), a solar system will prove cost effective. Solar systems designed for combined heating and cooling will utilize the collector year-around and thus usually will be more cost effective.

2.0 SOLAR SYSTEM COMPONENTS

2.1 Collectors

The collector is the most important and one of the most expensive parts of a solar heating system. It must be long-lived and well insulated, yet its cost must be minimized. Collectors of primary interest for space and water heating are of two basic types: liquid and air. Liquids may be water, an antifreeze mixture, or various hydrocarbon and silicone heat transfer oils. Air-type collectors use air as the collector fluid. The absorber plate is that part of the collector which absorbs the solar energy and converts it to thermal energy. A portion of the thermal energy is carried to the building or thermal storage unit by the fluid which circulates through passages in the absorber plate. The absorber plates can be made of metal, plastic, or rubber compounds. The metals commonly used in order of decreasing thermal conductivity are copper, aluminum, and steel. Plastics (polyolefins) and rubbers (ethylene propylene compounds) are relatively inexpensive, but due to their low thermal conductivity and their temperature limitations, they are suitable only for low temperature applications, such as heating swimming pool water or for use with water source heat pumps. Typical cross sections of solar collector types are shown in Figure 2-1.

Other major components of a solar collector include:

(a) Absorber plate coating - To enhance the heat transfer and protect the absorber plate.

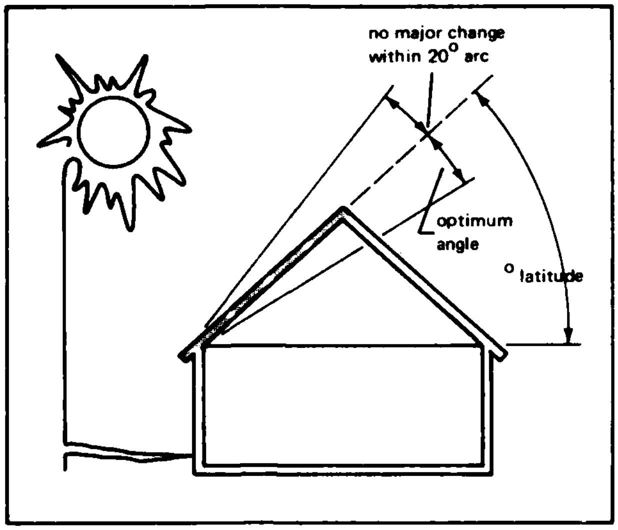


Figure 1-3. Collector tilt for domestic hot water (usually = latitude, but 10-degree variations either side of optimum are acceptable).

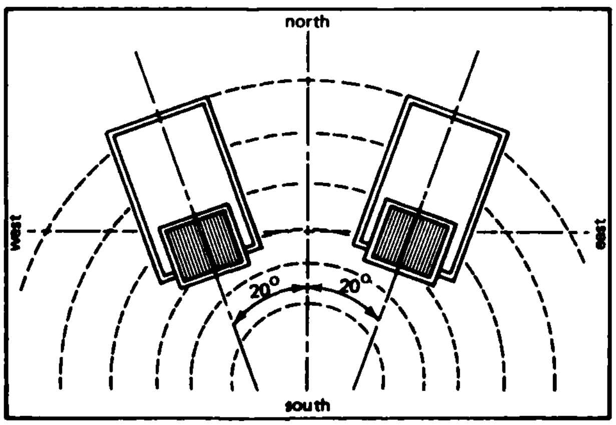


Figure 1-4. Collector orientation (optimum = true south, but 20-degree variations to either side are acceptable; local climate and collector type may influence orientation).

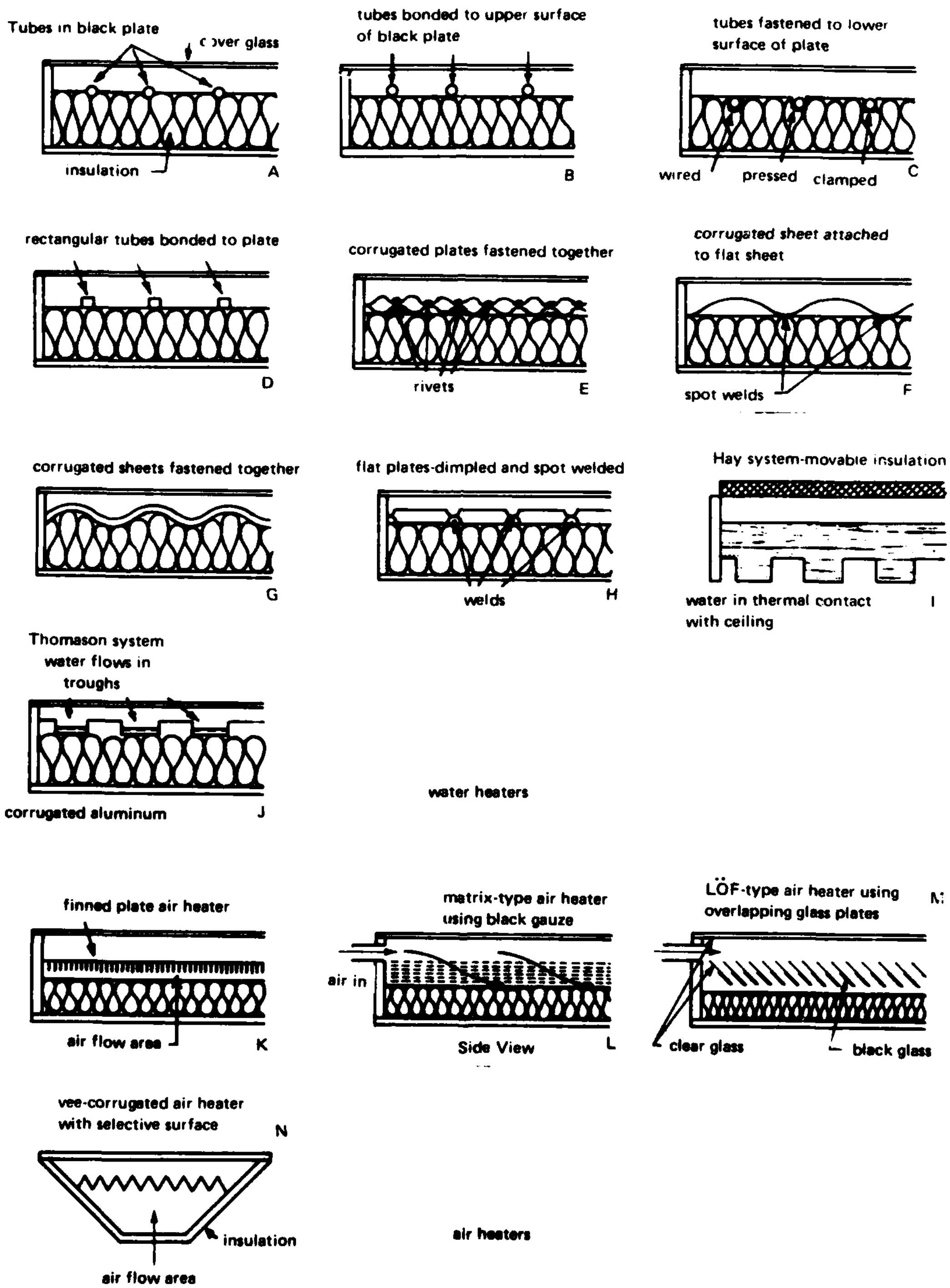


Figure 2-1. Types of solar heat collectors.

(b) One or more transparent covers - To reduce thermal losses by radiation (using the "greenhouse effect") and by convection (winds, etc.). Spacings are nominally 1/2 inch or more.

(c) Insulation - Two to six inches are used to reduce heat loss through the side and back of the absorber plate.

(d) Collector box or housing - To provide a rigid mounting to hold the components. Must be weatherproofed.

(e) Gaskets and seals - To insure a weathertight seal between components while allowing thermal expansion of the components. Normally these seals remain ductile to accomplish their purpose.

Flat-plate collectors are most suitable for low temperature applications such as domestic hot water and space heating. They collect both direct and diffuse radiation. It is not required that they track the sun, thus initial cost and maintenance are minimized. A properly designed flat-plate collector has a life expectancy of 10 to 25 years, or sometimes longer. All copper and glass systems currently exhibit the longest lives. Using softened water will help. Tubes should be 1/2 inch in diameter or greater for low pressure drop and longer life. The better the attachment of tube-to-plate (such as by soldering), the better the heat transfer, but the greater the manufacturing cost. Advances in collector cost reduction will probably be made in the direction of cheaper manufacturing processes. Some collectors not made from tube and sheet may not tolerate Domestic Hot Water (DHW) line pressures. Specifications for pressurized collector circuits should require collectors which will take proof test pressure equal to 150% of expected circuit pressure.

In hot climates, it is important to reduce roof heat load due to collector heat gain in summer; this can be accomplished by venting the space between collector plate and glazes with dampers or by covering the collectors. A normal amount of dirt and dust on the glass cover

will reduce heat collected by about 5%. Normal rainfall is usually sufficient to relieve this problem. Except for warm climates with high insolation ($I \geq 400$ L/day), two cover glasses may be optimum (see Section 2.1.3). In warm climates, one glass is optimum. Many plastics have an undesirable transparency to infrared radiation, to which glass is nearly opaque, so the desired "greenhouse effect" is not so pronounced with plastic materials as with glass. However, losses by radiation from the collector are small compared with convective losses due to wind; thus plastics can be employed to reduce breakage and cost, but with some loss in collector performance. Plastics with maximum opaqueness to infrared and maximum transparency to ultraviolet (UV) and visible radiation and with high resistance to UV degradation should be specified. Collector orientation should follow the guidelines given in Section 1.3.3. Collector sizing will be given in Section 3.0. The following sections give more detailed information on collector designs and components.

2.1.1 <u>Liquid and Air-Type Collectors</u>. Liquid and air type collectors each have some advantages which are summarized in Table 2-1 (Kimbell, 1978). Liquid types are more suited to domestic hot water, the collector area is usually smaller, and more information is available about liquid systems. Collectors for heating air do not require protection from freezing and have minimal corrosion problems, leaks do not cause serious problems, they may cost less per unit area, and are better suited to direct space heating for residences where duct-work is already present. Wherever this manual discusses liquid collectors, air collectors are included, and cost analyses apply equally to both. The design procedure for air collectors differs, however. Heat transfer oils used in liquid systems offer freeze protection and some corrosion protection, but they also require heat exchangers for heating domestic hot water, as do antifreeze-water mixtures.

2.1.2 <u>Selective Surfaces</u>. Some collectors are manufactured with a black coating which absorbs the high frequency incoming solar radiation very well and which emits low frequency infrared radiation poorly. This is a highly desirable combination of properties for a collector.

Table 2-1. Advantages and Disadvantages of Air and Liquid Heating Systems

Air	
Advantages	**Disadvantages**
Moderate cost. No freezing problems. Minor leaks of little consesequence. As air is used directly to heat the house, no temperature losses due to heat exchangers (devices which transfer heat from one fluid to another), when the system is used for space heat. No boiling or pressure problems.	Can only be used to heat homes; cannot presently be economically adapted to cooling. Large air ducts needed. Larger storage space needed for rocks. Heat exchangers needed if system is to be used to heat water.
Water or Liquid	
Advantages	**Disadvantages**
Holds and transfers heat well. Water can be used as storage. Can be used to both heat and cool homes. Compact storage and small conduits.	Leaking, freezing, and corrosion can be problems. Corrosion inhibitors needed with water when using steel or aluminum. There are liquids which are noncorrosive and nonelectrolytic; however, they are toxic and some of them are flammable. A separate collector loop using a nonfreezing fluid and a heat exchanger or, alternatively, a draining water or inhibited water system, are required to prevent freezing. In warm regions, where freezing is infrequent, electric warmers or recirculation can be used.

The absorptance should be 0.9 or higher and emittance may be 0.1 or lower. Such experimental coatings are expensive and are approximately equal in effect to one cover glass. Thus, a selective coating plus one cover glass may be expected to be about equal in efficiency to a collector with two cover glasses and a flat black painted surface. Electroplated black nickel, chrome, copper oxide or anodized aluminum are common types of selective coatings. Cost of such coatings may now be greater than an extra sheet of glass, but in the future costs will probably be less than the glass. The stability of black nickel, chrome and aluminum in the presence of moisture has not yet been proven. Long-term stability in the presence of moisture or other expected environmental factors (salt air, etc.) must be included in specifications for selective surfaces. Table 2-2 is a summary of absorber coatings both selective and nonselective.

2.1.3 Collector Covers (Glazes). The transparent covers serve to admit solar radiation to the absorber while reducing convection and radiation heat losses from the collector. The covers also protect the absorber from dirt, rain, and other environmental contaminants.

The material used for covers include glass and/or plastic sheets. Glass is most commonly used because of its superior optical properties and durability. Standard plate glass reflects about 8% and absorbs about 6% of normal incident solar radiation, resulting in a transmissivity of about 86%. Yet it is essentially opaque to long-wave thermal radiation from the absorber. Transmission of solar radiation into the collector can be increased by minimizing the reflectance and the absorptance of the glass covers. Absorptance of solar radiation can be decreased with the use of thinner tempered glass and by using glass that has a low iron content. Although glass is subject to impact damage and is more expensive than plastic, it does not degrade in sunlight or at high collector temperatures, and is generally considered to be more durable than plastic. Impact damage may be reduced with the use of tempered glass and small collector widths. Also 1/2-inch wire mesh may be hung over glass covers for protection, but the effective absorber area will be reduced by approximately 15%. In general, screens are not recommended.

Table 2-2. Characteristics of Absorber Coatings (U.S. Dept HUD, 1977)

[Selective Coatings - $\alpha/\varepsilon > 2$; Non-Selective Coatings - $\alpha/\varepsilon \sim 1$]

Property/ Material	Absorptance,[a] α	Emittance, ε	$\frac{\alpha}{\varepsilon}$	Breakdown Temperature, °F (°C)	Comments
Black Chrome	0.87-0.93	0.1	~9		
Alkyd Enamel	0.9	0.9	1		Durability limited at high temperatures.
Black Acrylic Paint	0.92-0.97	0.84-0.90	~1		
Black Inorganic Paint	0.89-0.96	0.86-0.93	~1		
Black Silicone Paint	0.86-0.94	0.83-0.89	~1		Silicone binder.
PbS/Silicone Paint	0.94	0.4	2.5	662 (350)	Has a high emittance for thicknesses >10 µm
Flat Black Paint	0.95-0.98	0.89-0.97	~1		
Ceramic Enamel	0.9	0.5	1.8		Stable at high temperatures.
Black Zinc	0.9	0.1	9		
Copper Oxide Over Aluminum	0.93	0.11	8.5	392 (200)	
Black Copper Over Copper	0.85-0.90	0.08-0.12	7-11	842 (450)	Patinates with moisture.

(continued)

Table 2-2. Continued

[Selective Coatings - $\alpha/\varepsilon > 2$; Non-Selective Coatings - $\alpha/\varepsilon \sim 1$]

Property/ Material	Absorptance,[a] α	Emittance, ε	$\frac{\alpha}{\varepsilon}$	Breakdown Temperature, °F (°C)	Comments
Black Chrome Over Nickel	0.92-0.94	0.07-0.12	8-13	842 (450)	Stable at high temperatures.
Black Nickel Over Nickel	0.93	0.06	15	842 (450)	May be influenced by moisture at elevated temperatures.
Ni-Zn-S Over Nickel	0.96	0.07	14	536 (280)	
Black Iron Over Steel	0.90	0.10	9		

[a]Dependent on thickness and vehicle-to-binder ratio.

Most plastic covers transmit the solar spectrum as well or better than glass glazing. Unfortunately, they transmit infrared radiation well also, increasing radiation losses from the collector. Table 2-3 compares the different characteristics of glass and plastic covers (Montgomery, 1978).

Although resistant to impact damage, plastics generally degrade in sunlight and are limited as to the temperatures they can sustain without undergoing serious deformation. In general, acrylic is the most UV resistant and polycarbonate offers good impact and high temperature properties. Teflon FEP film has good transmittance and high temperature properties, but is limited in strength. Some collectors using plastic covers are designed to have stagnation temperatures no higher than 200°-275°F. However, plastic covers have been developed to withstand 400°F. The manufacturer should be consulted.

Each additional cover, whether it be glass or plastic, reduces convection heat losses but results in added expense and less solar radiation transmitted to the absorber. Most commercially available collectors come with one or two covers. The decision to use one or two covers depends on the type of absorber coating, the required collection temperatures, average ambient air temperature, the local wind conditions, and of course, the cost of the covers.

As stated in Section 2.1.2, the use of a selective surface is about equal to using one additional cover. Thus for most cases, only one glass cover is needed if the absorber has a selective coating. In fact, one study indicated that winter performance was actually reduced by the use of two glass covers with a selective surface compared to one cover with the selective surface.

Two covers are generally recommended for use in Northern climates where winter ambient air temperatures are low. For flat-plate collectors used mostly for winter heating, one rule of thumb is to use one glass cover where average winter air is greater than 45°F, and two glass covers in colder climates. Table 2-4 gives some approximations in the selection of collector covers.

Table 2-3. A Comparison of Various Materials Used for Collector Covers

Glazing Type	Solar (Shortwave) Transmittance, %	Infrared (Longwave) Transmittance, %	Index of Refraction	Weatherability[a] and Durability
White Crystal Glass	91.5	2	1.50	Excellent
Low-Iron Tempered Glass	87.5	2	1.51	Excellent
Low-Iron Sheet Glass	87.5	2	1.51	Excellent
Tempered Float Glass	84.3	2	1.52	Excellent
Fiberglass	77 to 87	0.1 to 0.3	1.54	Fair to Good
Sheet Acrylic	80 to 90	2	1.49	Average to Good
Sheet Polycarbonate	73 to 84	2	1.59	Poor to Fair
FEP Teflon	90 to 92	25 to 26	1.34	Fair to Good
Polyester Film	80 to 87	20 to 21	1.64 to 1.67	Fair to Good

[a]Durability of many plastics is still to be determined by field use and may change the ratings up or down.

Table 2-4. Guide to Selection of Number of Transparent Cover Plates

Collection Temperature Above Ambient Temperature (t_c-t_a)	Typical Applications	Optimum Number of Cover Plates	
		Black-Painted Absorber $\varepsilon = 0.9$ or 0.95	Selective Absorber $\varepsilon = 0.2$
-5°C to +5°C (-10°F to +10°F)	Heat source for heat pump Heating of swimming pools in summer Air heating for drying	none	none
5°C to 35°C (10°F to 60°F)	Domestic water heating Heating of swimming pools in winter Air heating for drying Solar distillation Space heating in non-freezing climates	1	1
35°C to 55°C (60°F to 100°F)	Winter water heating Winter space heating	2	1
55°C to 80°C (100°F to 150°F)	Summer air conditioning Steam production in summer Refrigeration Cooking by boiling	3	2

2.1.4 Collector Insulation. Insulation behind and to the side of the absorber serves to reduce conduction losses. Usually, this insulation consists of 2-6 inches of high-temperature fiberglass batting or semi-rigid board or even mineral wool. Styrofoam and urethane foams are usually not used because they may deform at high temperatures or give off gases (which may be toxic). The insulation should be separated from the absorber plate by 1/2 to 3/4 inch and have a reflective foil facing the absorber plate. If fiberglass insulation is used, it should not be typical construction grade which contains phenolic binders that may "outgas" at the stagnation temperature of the collector. In all cases, specifications should call for insulations that have a low thermal expansion coefficient, do not melt or outgas at collector stagnation temperatures (300°-400°F), and (whenever possible) contain reflective foil to reflect thermal radiation back to the absorber.

2.1.5 Collector Housings. The housing or collector box serves to

(1) Support the collector components

(2) Protect the absorber and insulation from the environments

(3) Reduce convection and conduction losses from the absorber

Many housing designs are available on the market. They are constructed of metals, wood, plastics, concrete, and other materials. The most commonly used materials are aluminum, galvanized sheet metal, fiberglass laminates, high temperature thermoplastics, and wood (Montgomery, 1978).

All structural materials are suitable if properly used. However, most commercially available housings consist of a galvanized sheet metal box with an anodized aluminum frame which fits on top of the box.

Some housings are designed to be integrated directly into the roof or wall structure, thus reducing construction costs.

Since field labor is expensive, the collector housing should be designed such that the collector units can be quickly secured in place and connected to the external piping. Provisions should also be made

for easy replacement of broken glass covers. The absorber plate should be mounted so as to be thermally isolated as much as possible from the housing.

2.1.6 Collector Gaskets and Sealants. Gaskets and sealants must be carefully selected if a collector is to have a long life. Generally, the housing and the glazing have different rates of thermal expansion. Gaskets and sealants form the flexible interface between the two components and seal out moisture and other contaminants; if they fail, moisture will fog the glazing and may possibly damage the absorber coating and the insulation. These problems can drastically reduce the thermal performance of the collector.

Two suitable sealing methods are shown in Figures 2-2 and 2-3 (Montgomery, 1978). The gaskets provide flexible support and the primary weather sealant insures against moisture leakage. Dessicants are sometimes placed between the two glazings to absorb any moisture that may remain after cover installation.

When selecting collector gaskets and sealants, certain material requirements must be kept in mind. The gaskets and seals must

(1) Withstand significant expansion and contraction without destruction

(2) Adhere effectively to all surfaces

(3) Resist ultraviolet degradation

(4) Resist outdoor weathering

(5) Not harden or become brittle

(6) Withstand temperature cycling from -30° to 400°F

Both EPDM and silicone rubbers have been found adequate for use as gasket materials. Silicone sealants have exceptional weathering resistance and have received widespread use for many years.

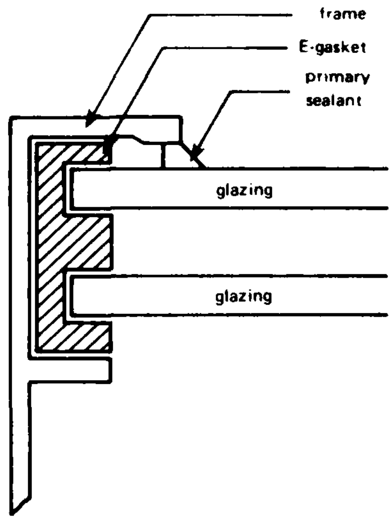


Figure 2-2. Single gasket seal for double glazing.

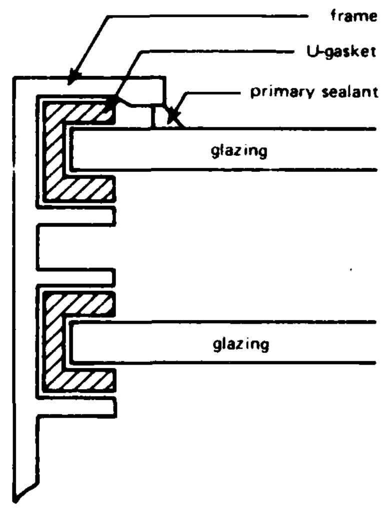


Figure 2-3. Typical sealing method for single or double glazing.

2.1.7 <u>Collector Fluids - Corrosion and Freeze Protection</u>. The choice of which collector fluid to use is important because this is the life-blood of the system. The cheapest, most readily obtainable, and thermally efficient fluid to use is ordinary water. However water suffers from two serious drawbacks - it freezes and it can cause corrosion. Therefore, the choice of collector fluid is closely linked to the type of solar system, the choice of components, future maintenance, and several other factors which will be discussed in this section.

Implicit in this discussion is the use of a fluid other than air as the collector fluid. As explained in Table 2-1 an air solar system does not suffer from corrosion or freezing effects, but its low density and heat capacity requires the use of fans and large ducts, large storage volumes, and is generally not suitable for domestic water heating. The remainder of this section applies to liquid solar heating systems.

A list of standards has been prepared for heat transfer fluids and can be found in the reference, "Intermediate Minimum Property Standards" (see Section 1.2). Generally the standards state the heat transfer fluid must be noncorrosive, nonflammable and stable with temperature and time. If the fluid is toxic it may be used only in systems specially designed for it as will be explained later.

If there is no danger of freezing and the collector loop consists of all copper flow passages, then ordinary water would be the choice for collector fluid. If freezing conditions are encountered, there are a number of designs that should be considered before it is decided to use a heat transfer oil or antifreeze mixture. These freeze protection schemes are summarized here using Figure 2-4 as the basic open loop type collector circuit.

(a) Drain Down Method - The water in the collector is drained into the storage tank when temperatures in the collector approach freezing. This scheme requires automatic valves to dump the water and purge air from the system. Often a larger pump will be required to overcome the system head and re-prime the collectors. A way to avoid automatic (solenoid) valves is to drain the collectors whenever the pump shuts off. This still requires a larger pump. Heat exchangers may be required to separate potable water from nonpotable water.

(b) Heat Tapes - Electric resistance heat tapes are thermostatically activated to heat the water. This scheme requires extra energy and is not completely reliable. Insertion of heat tapes into preconstructed collectors may be difficult.

(c) Recirculation Method - In this method the control system of Figure 2-4 merely turns on the pump if freezing approaches. In this way, warm water from storage circulates through the collectors until the freezing condition is over. The only extra component needed is a freeze sensor on the collector which is a minimum cost item ($5-$10). However, by circulating heated water, the capacity of storage decreases and less is available the following day. This method is probably the most reliable of the three since it does not depend on additional electrical valves or heating tape.

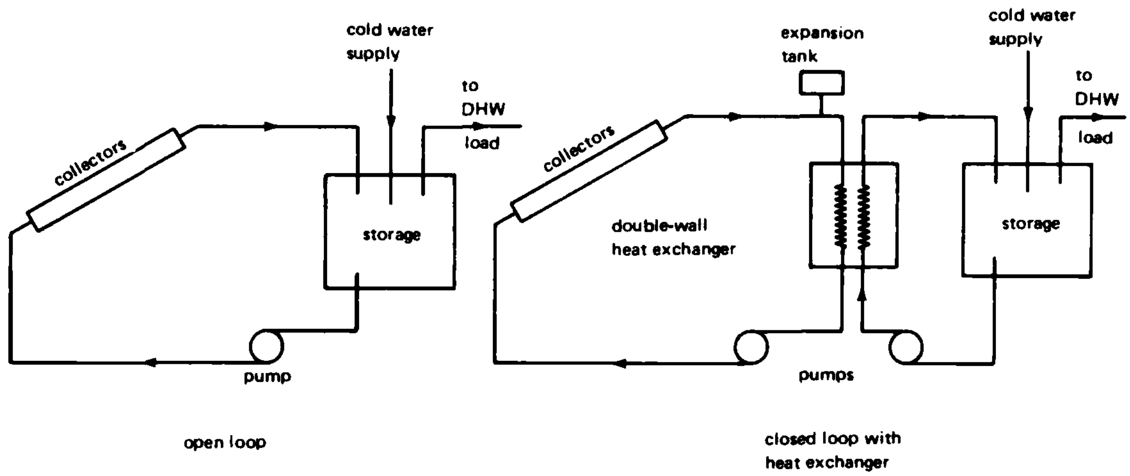
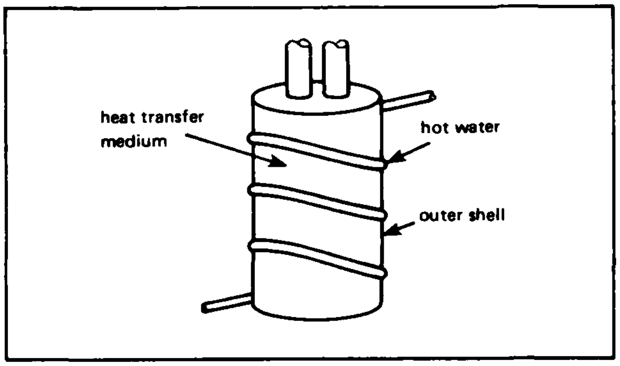


Figure 2-4. Typical configurations for solar water heater systems.

If the preceding methods are not acceptable or if the choice of water is not acceptable due to concern about corrosion, then a heat transfer fluid must be used. The heat transfer fluid must be used with a heat exchanger in a "closed-loop" configuration as shown in Figure 2-4.

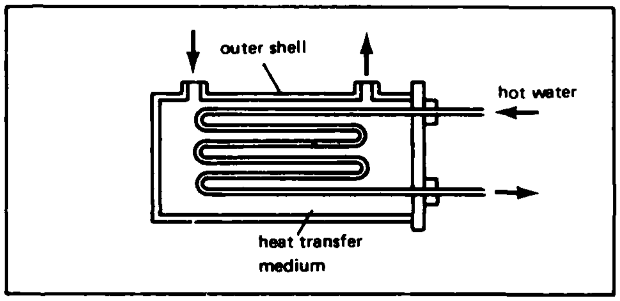
The configuration shown in Figure 2-4 will be from 10%-25% less efficient due to the temperature penalty associated with the heat exchanger and the low specific heat of the heat transfer fluid as compared to water. Note an additional pump is also required. If the heat transfer fluid is toxic (such as antifreeze) then a double-walled heat exchanger must be used for protection. The different types of heat exchangers are explained in Figure 2-5 (National Solar Heating and Cooling Info Ctr, 1979).

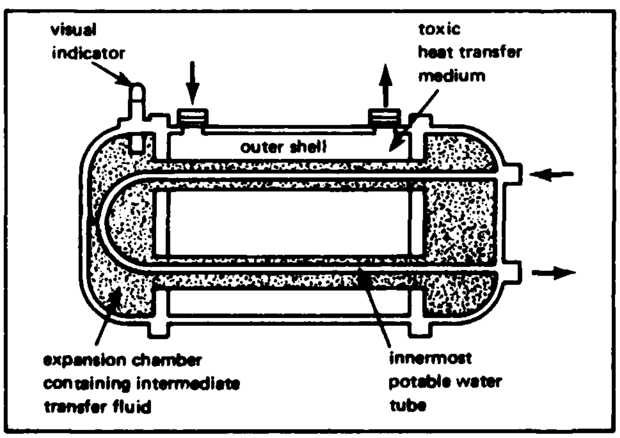
It is difficult to estimate the most cost effective freeze protection method. Some studies have shown that for many areas in the U.S., the recirculation method is best particularly where freezing days are few in number. It tends to have the lowest capital cost and energy use cost. However all the methods except heat transfer fluids rely on the presence of electricity to operate. A simultaneous electrical failure and freezing condition would result in potential failure of the systems. Therefore, the absolute safest system would be the nonfreezing heat



Double Wall. Another method of providing a double separation between the transfer medium and the potable water supply consists of tubing or a plate coil wrapped around and bonded to a tank. The potable water is heated as it circulates through the coil or through the tank. When this method is used, the tubing coil must be adequately insulated to reduce heat losses.

Shell and Tube. This type of heat exchanger is used to transfer heat from a circulating transfer medium to another medium used in storage or in distribution. Shell and tube heat exchangers consist of an outer casing or shell surrounding a bundle of tubes. The water to be heated is normally circulated in the tubes and the hot liquid is circulated in the shell. Tubes are usually metal such as steel, copper or stainless steel. A single shell and tube heat exchanger cannot be used for heat transfer from a toxic liquid to potable water because double separation is not provided and the toxic liquid may enter the potable water supply in a case of tube failure.





Shell and Double Tube. This type of heat exchanger is similar to the previous one except that a secondary chamber is located within the shell to surround the potable water tube. The heated toxic liquid than circulates inside the shell but around this second tube. An intermediary nontoxic heat transfer liquid is then located between the two tube circuits. As the toxic heat transfer medium circulates through the shell, the intermediary liquid is heated, which in turn heats the potable water supply circulating through the innermost tube. This heat exchanger can be equipped with a sight glass to detect leaks by a change in color—toxic liquid often contains a dye—or by a change in the liquid level in the intermediary chamber, which would indicate a failure in either the outer shell or intermediary tube lining.

Figure 2-5. Heat exchangers for solar water heating systems.

transfer fluids and these might be considered for the very cold parts of the country (Boston, Chicago, etc.). Each potential project should be considered individually using local weather criteria, freeze protection capital costs, additional energy to run the system, reliability, maintenance, and type of system as the criteria. Often a detailed computer simulation would be required to choose. However, any of the methods will provide some degree of protection. If heat transfer fluids are selected for corrosion or freeze protection, the following paragraphs discuss pertinent criteria.

Before heat transfer fluids are discussed, a review of basic corrosion theory is in order. The two types of corrosion which cause the most damage in solar systems are galvanic and pitting corrosion (Eyre, 1978). Galvanic corrosion is a type of corrosion which is caused by an electrochemical reaction between two or more different metals in contact with each other. A chemical reaction between the metals causes a small electrical current which erodes material from one of the metals. Solar energy systems generally contain a number of different metals such as aluminum, copper, brass, tin, and steel. This makes the solar system a prime candidate for galvanic corrosion. If the dissimilar metals are physically joined or if they are contacted by a common storage or heat-transfer fluid, the possibility of galvanic corrosion becomes much greater.

Pitting corrosion is a highly localized form of corrosion resulting in deep penetration at only a few spots. It is one of the most destructive forms of corrosion because it causes equipment to fail by perforation with only a very small weight loss. When heavy metal ions such as iron or copper plate out on a more anodic metal such as aluminum, a small local galvanic cell can be formed. This corrosion spot or "pit" usually grows downward in the direction of gravity. Pits can occur on vertical surfaces, although this is not as frequent. The corrosion pits may require an extended period (months to years) to form, but once started they may penetrate the metal quite rapidly.

Heavy metal ions can either come as a natural impurity in a water-mixture heat transfer fluid or from corrosion of other metal parts of the solar system.

Pitting corrosion has the same mechanism (concentration cell) as crevice corrosion thus it can also be aggravated by the presence of chloride or other chemicals which can be part of the water mixture or a contaminant from solder fluxes. Aluminum is very susceptible to pitting corrosion, while copper generally is not.

There are several preventive measures which will eliminate or at least minimize galvanic and pitting corrosion in collector systems which use an aqueous collector fluid. Galvanic corrosion is prevented by using nonmetallic connections between dissimilar metals. Pitting corrosion is essentially eliminated if copper absorber plates are used. Corrosion inhibitors can minimize pitting corrosion in aluminum absorbers.

The types of heat transfer fluids available may be divided into two categories, nonaqueous and aqueous. Silicones and hydrocarbon oils make up the nonaqueous group, while the aqueous heat transfer fluids include untreated potable (tap) water, inhibited-distilled water, and inhibited glycol/water mixtures. The potable tap water and inhibited distilled water do not, of course, offer freeze protection. Table 2-5 shows characteristics of some of the most common heat transfer fluids.

2.1.7.1 Silicone Fluids: Silicone heat transfer fluids have many favorable properties which make them prime candidates for collector fluids. They do not freeze, boil, or degrade. They do not corrode common metals, including aluminum. They have excellent stability in solar systems stagnating under 400°F. Silicone fluids are also virtually nontoxic and have high flash and fire points. Current evidence indicates that silicone fluids should last the life of a closed-loop collector system with stagnation temperatures under 350°-400°F. The flash point is fairly high, 450°F, but since the HUD standards state that heat transfer fluids must not be used in systems whose maximum stagnation temperature is less than 100°F lower than the fluid's flash point, this limits most silicone oils to systems with a maximum temperature of 350°F or less. Also silicones do not form sludge or scale, so system performance does not decrease with time.

The main drawback of silicone fluids is their cost. Currently silicone fluid costs about \$20-\$25 per gallon. Thus the cost of the 20 to 30 gallons of collector fluid required for a typical 500 ft^2 collector system becomes considerable. As with hydrocarbon oils, the lower heat capacity and higher viscosity of silicone fluid requires larger diameter and more expensive piping. Due to the higher viscosity, larger pumps will be required and subsequent higher pumping costs. One other problem with silicone fluids is the seepage of fluid at pipe joints. This problem can be prevented by proper piping installation and by pressurizing the system with air to test for leaks. There have also been reports of seepage past the mechanical seals of circulating pumps.

Silicones have the advantage of lasting the life of the system with little maintenance. While this helps minimize operating expenses, the initial cost of silicones is markedly higher than that of other available heat transfer fluids. However, the high initial cost of silicone heat transfer fluid may be less than the savings that result from minimum maintenance and no replacement of collector fluid. The use of silicone fluid allows aluminum absorbers to be used without fear of corrosion. The savings gained from the use of aluminum absorbers as opposed to copper absorbers could be substantial.

2.1.7.2 Hydrocarbons: Hydrocarbon oils, like silicones, also give a long service life, but cost less. They are relatively noncorrosive, nonvolatile, environmentally safe, and most are nontoxic. They are designed for use in systems with lower operating temperatures, since some brands break down at higher temperatures to form sludge and corrosive organic acids. Typical closed-cup flashpoints run from 300°F to 420°F, but the fluids with higher flashpoints have a higher viscosity. The HUD bulletin on minimum property standards for solar heating systems recommends a closed-cup flashpoint 100°F higher than maximum expected collector temperatures.

Unsaturated hydrocarbons are also subject to rapid oxidation if exposed to air, necessitating the use of oxygen scavengers. Some hydrocarbons thicken at low temperatures and the resultant higher viscosity can cause pumping problems.

Table 2-5. Heat Transfer Fluids (Solar Engineering, 1978)

[cs = centistokes; cps = centipoise; all degrees are Fahrenheit]

Company	Useful Temperature Range (°F)	Specific Heat (Btu/lb/°F)	Viscosity	Specific Gravity	Toxicity	Flash Point	Other Features/Specifications
Bray Oil Co., Inc. **Los Angeles, CA**							
Synthetic hydrocarbon Brayco 888	-100 to 550	0.55 (approx.)	6.3 cs at 100° 1.9 cs at 210°	0.8 at 60°/60°	low	325° (med.)	Formulated for copper and aluminum systems. Noncorrosive and nonpoisonous.
Dow Chemical, U.S.A. **Midland, MI**							
Dowfrost inhibited propylene glycol	-28 to 300	0.35 at 80° (50% water solution)	60 cps at 0° (50% water solution)	1.05	low	214° (low)	Used as a secondary coolant fluid.
Dowtherm SR-1, inhibited ethylene glycol	-40 to 300	0.87 at 200° (50% SR-1 by weight)	0.9 cps at 200° (50% SR 1 by weight)	1.13	high	250° (low)	Provides corrosion protection for all common metals.
Dowtherm J, alkylated aromatic fluid	100 to 575 with 10-20 psig	0.5 at 200°	0.7 cps at 100° 0.35 cps at 200°	0.87	low	145° (low)	Resists both thermal degradation and oxidation.
Dowtherm HP	15 to 550	0.640				420° (med.)	
Dow Corning Corp. **Midland, MI**							
Dow Corning Q2-1132 silicone heat transfer liquid	-50 to 450	0.37 at 104° 0.42 to 392°	20 cs at 77° 7 cs at 210°	0.946 at 77°	low	450° (high)	Water clear liquid. Essentially noncorrosive. Negligible thermal degradation at 400°.
Slytherm 444, silicone heat transfer fluid	50 to 300	0.42 at 392°	20 cs at 77°	0.98 at 300°	low	450° (high)	Similar to Q2-1132.
Drew Chemical Corp. **Booton, NJ**							
Drewsol heat transfer fluid	28.5 to 230	0.87 (avg.)	8.8 cps at 77°	1.136 (avg.)	none	none	Inhibits corrosion. Will not deteriorate asphalt.

(continued)

Table 2-5. Continued.

[cs = centistokes; cps = centipoise; all degrees are Fahrenheit]

Company	Useful Temperature Range (^{o}F)	Specific Heat (Btu/lb/^{o}F)	Viscosity	Specific Gravity	Toxicity	Flash Point	Other Features/Specifications
Exxon							
Caloria HT 43 petroleum base with parafinic stock	15 to 600	0.51	30.9 cs at 100^{o}	1.0		420^{o} (med.)	
Mark Enterprises, Inc. Woodbridge, CT							
H-30C: synthetic hydrocarbon heat transfer fluid (blue)	-40 to 620	0.50	180 cps at 32^{o} 18 cps at 100^{o} 4.3 cps at 200^{o}	0.845	none	360^{o} (med.)	Noncorrosive. Will not attack rubber. High thermal conductivity.
H-30: synthetic hydrocarbon heat transfer fluid (green)	-40 to 560	0.57	180 cps at 32^{o} 20 cps at 100^{o} 8.2 cps at 200^{o}	0.835	none	310^{o} (med.)	Noncorrosive. Will not attack rubber. High thermal conductivity.
Mobil							
Mobiltherm 600 refined oils	5 to 600	0.56	-	1.0		350^{o} (med.)	Highly aromatic.
Monsanto Industrial Chemicals Co. St. Louis, MO							
Therminol 44, 55, 60 66 and 88 synthetic based fluids	-60 to 800	0.32 to 0.72	4,000 to 0.23 cs	0.79 to 1.04 over range	low	310^{o} to 405^{o} (med.)	Wide range of applications.
Nuclear Technology Corp. Amston, CT							
Nutek 800 Series:							
• Ethylene-glycol	-30 to 230	0.80 to 0.95	3.4 cps at 100^{o} 0.6 cps at 200^{o}	1.03 to 1.07			Nutek 800 series are designed for use in aluminum solar collector panels where corrosion and scaling protection is required.
• Propylene-glycol and water-based compounds	32 to 212	1.0	0.7 cps at 100^{o} 0.3 cps at 200^{o}	1.0			

(continued)

Table 2-5. Continued

[cs = centistokes; cps = centipoise; all degrees are Fahrenheit]

Company	Useful Temperature Range (^{o}F)	Specific Heat (Btu/lb/^{o}F)	Viscosity	Specific Gravity	Toxicity	Flash Point	Other Features/Specifications
Resource Technology Corp. New Britain, CT							
Sun-Temp: nonaqueous heat transfer fluid	-40 to 500	0.56	89 cps at 33° 2.1 cps at 212°	0.9 at 72°	none	380° (med.)	Noncorrosive.
Shell							
Hydrocarbon oil Thermia C	20 to 550	0.46	62.9 cs at 100°	–	–	455° (high)	–
Union Carbide Corp. Tarrytown, NY							
Inhibited ethylene glycol	-90 to 240	0.79 at 190°	60 cps at -10° 1.5 cps at 180°	1.1 at 60°/60°	high	low	Formulated for use in multi-metal systems. All values at 70% volume in water.
UCON 30	–	0.458	65.6 cs at 100°	-	-	535° (high)	-
Uniroyal Chemical Division of Uniroyal, Inc. Naugatuck, CT							
Synthetic polyalphaolefins	-40 to 400	0.50	10 cs at 200°	0.83	none	400° (med.)	Relatively nontoxic and non-corrosive. Unusually low pour point and volatility.

Newer hydrocarbons are being developed which do not harm rubber or materials of construction, since this has been a problem with hydrocarbons. In general, they cannot be used with copper, as it serves as a catalyst to fluid decomposition. The thermal conductivity of hydrocarbons is lower than that of water, although the performance of some brands is much better than others.

The cost of typical hydrocarbon heat transfer oils vary from about $3/gal to $7/gal. A typical liquid collector of 500 ft^2 plus the piping to and from storage will require from 20 to 30 gallons of collector fluid. The lower heat capacity and higher viscosity of these oils will also require larger diameter pipe, increasing materials costs further. If hydrocarbon fluids are used, the additional capital cost should be compared with expected savings due to lower maintenance costs. The use of aluminum absorbers rather than copper absorbers will also result in substantial savings.

2.1.7.3 Distilled Water: Distilled water has been suggested for use in solar collectors since it avoids some of the problems of untreated potable water. First, since the distillation process removes contaminants such as chlorides and heavy metal ions, the problem of galvanic corrosion, though not completely eliminated, should be alleviated. However, distilled water is still subject to freezing and boiling. For this reason, an anti-freeze/anti-boil agent such as ethylene glycol is often added.

2.1.7.4 Water/Antifreeze: Nonfreezing liquids can also be used to provide freeze protection. These fluids are circulated in a closed loop with a double wall heat exchanger between the collector loop and the storage tank (see Figure 2-5).

Water/antifreeze solutions are most commonly used because they are not overly expensive. They range from $3-$4 per gallon including inhibitors. Ethylene and propylene glycol are the two most commonly used antifreezes. A 50-50 water/glycol solution will provide freeze protection down to about -30°F, and will also raise the boiling point to about 230°F.

The use of water/glycol solution presents an additional corrosion problem. At high temperatures glycols may break down to form glycolic acid. This acid corrodes most all metals including copper, aluminum, and steel. The rate of glycol decomposition at different temperatures is still a subject of uncertainty. The decomposition rate of glycol varies according to the degree of aeration and the service life of the solution. Most water/glycol solutions require periodic monitoring of the pH level and the corrosion inhibitors. If these solutions are used in the collector loop, the seller should specify the expected life of the solution and the amount of monitoring required. The cost of periodic fluid replacement and monitoring should be considered in the economic analysis.

Since glycol-water mixtures do require a lot of maintenance (and since homeowners can be quite negligent) it is recommended that glycols not be used in home solar heating and domestic hot water systems, and that glycol-water solutions be reserved for use in large-scale installations which have regular maintenance schedules and where the high cost of silicone oils would be prohibitive.

2.1.8 Collector Connections. Water flow through nonhorizontal collectors should always be against gravity, except in trickle-type collectors. Usually this means water inlet to the collector at the bottom, and outlet at top. Care must be taken so that equal flow goes to all tubes. If manifold ΔP is large, then center tubes will get little flow. The design most usually used is one in which the collectors are connected in parallel. This results in low pressure drop and high efficiency of each collector. A series hookup results in the highest temperature and the highest pressure drop but lowest collector efficiency. Higher temperatures than in the parallel arrangement may be obtained with parallel-series connections, but at the expense of reduced efficiency and greater cost. These high temperatures are not usually required for hot water and space heating. Figure 2-6 shows different connection configurations. Very large installations may merit computer simulations to optimize the various connections of each stage.

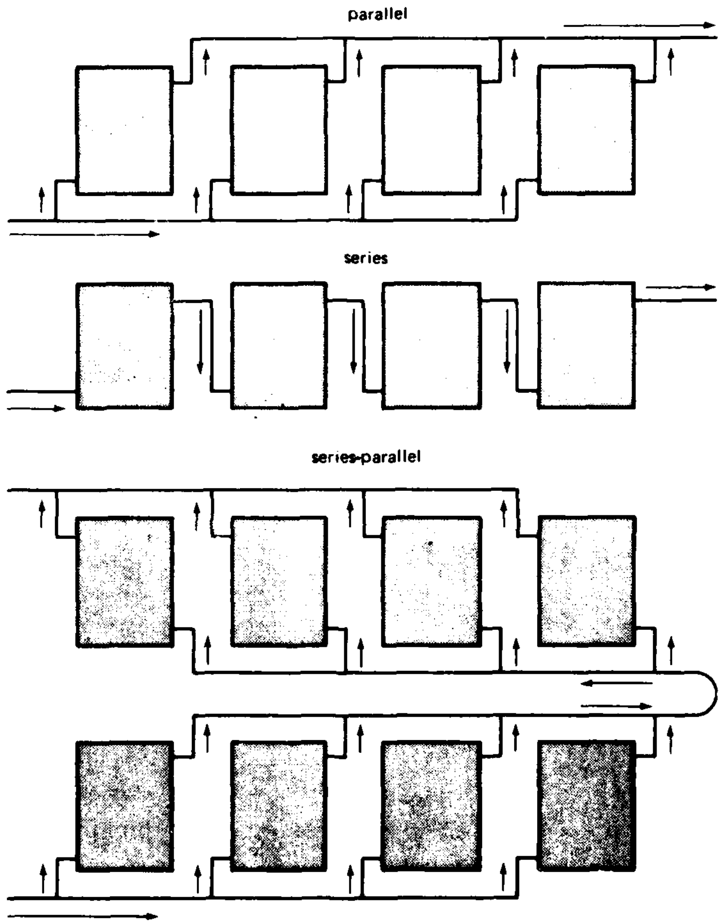


Figure 2-6. Connection schemes for solar heating systems.

2.1.9 Collector Efficiency and Heat Losses. In the preceding sections, many details as to the construction and choice of components of a solar collector have been given. All of these features contribute to how well a collector will perform or how efficient it will be. Solar collectors, depending on their construction and materials, suffer from several kinds of heat losses. They can lose heat by convection of wind blowing over their top and bottom surfaces. As the collector temperature increases above the temperature of the surrounding air, the radiation heat losses increase. This results in lower heat collected (lower efficiency) at higher collector temperatures. Heat can be lost by conduction from the back and sides of a collector. To evaluate the effects of all these parameters individually would involve detailed and difficult calculations.

Fortunately, collector performance can be compared much more easily by a single graph depicting collector efficiency versus the parameter $\Delta T/I$. Collector efficiency is defined as the ratio of the heat collected to the insolation (I) falling on the surface of the collector. Also:

$$\Delta T = T_i - T_a$$

where T_i = temperature of fluid entering collector (inlet)
T_a = ambient air temperature

Figure 2-7 gives the efficiency of some typical flat plate solar collectors. The most efficient solar collector would convert 100% of the sun's energy falling on it to usable heat. As shown in Figure 2-7, this is impossible so the designer looks for a collector that converts the greatest percentage of solar energy to heat, at the required temperature, and at the lowest cost.

It is important that each collector be tested according to an exacting standard. The early standard for testing solar collectors was NBSIR 74-635 published by the National Bureau of Standards (Hill and Kusada, 1974). This is the standard the previous edition of this report used to report collector efficiencies. Subsequently, the American Society of Heating, Refrigerating, and Air Conditioning Engineers, Inc.

Note: The units of $\Delta T/I$-^{o}F-day/Langley are for a six hour sunlight day, not a 24 hour day. This must be observed when ^{o}F-hr ft^2/Btu data are converted to be plotted hereon.

No.	Description	Ref Table 2-6
1	Single glass-alum plate	1
2	Single glass-copper plate-selective	22
3	Double glass-copper plate-selective	25
4	Single FRP-copper tube on alum plate	14
5	Unglazed-propylene plate	44
6	Single glass-copper-selective-evacuated	40

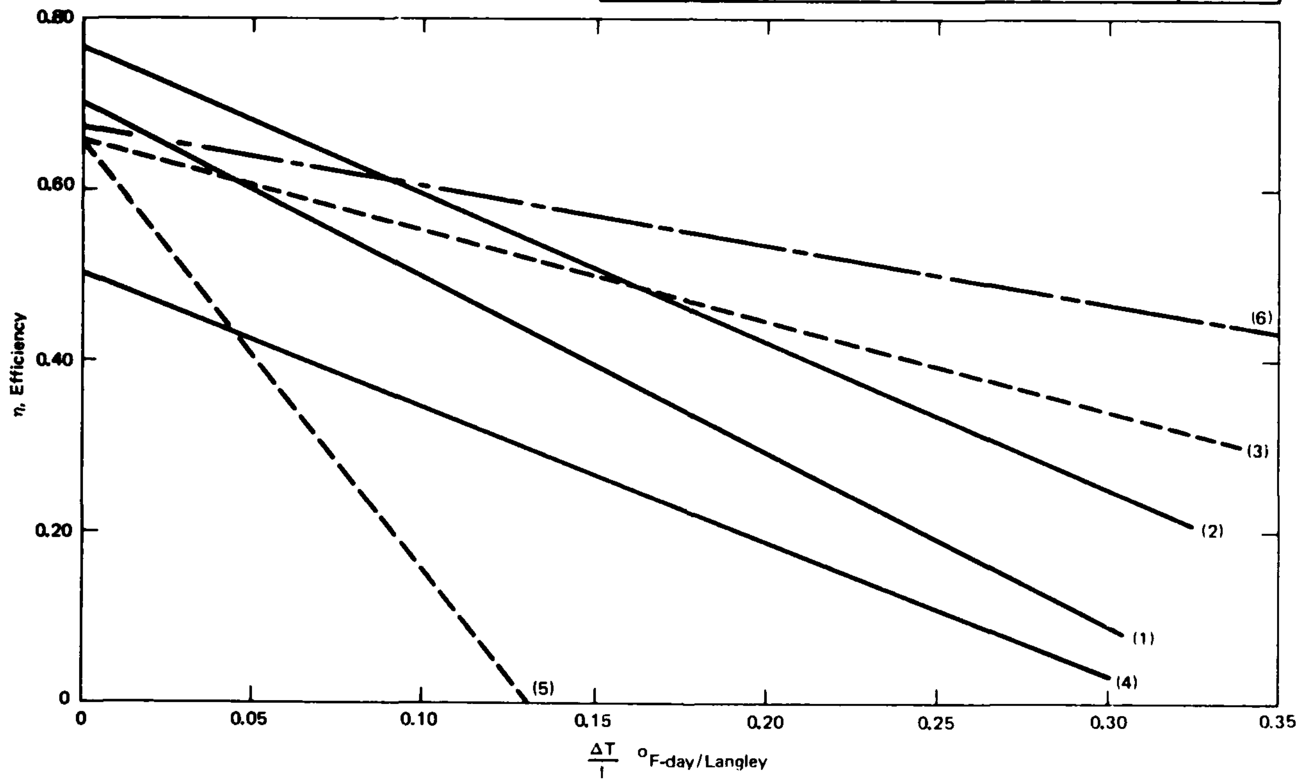


Figure 2-7. Typical solar heater collector efficiencies.

(ASHRAE) sponsored the development of a uniform method of testing solar collectors to form the preliminary standard 93-P and finally the version in use today, Standard 93-77, "Methods of Testing to Determine the Thermal Performance of Solar Collectors." This method uses the Hottel-Whillier equation and is generally accepted in the solar industry.

The differences between the NBS and the ASHRAE standard are as follows:

1. ASHRAE requires the use of gross collector area rather than aperture or net area used in NBS.

2. ASHRAE uses the collector inlet temperature as one of its parameters whereas NBS uses the average absorber plate temperature defined as the sum of the inlet and outlet temperatures divided by two.

In general, the NBS standard will give higher collector efficiencies, possibly 5%-10%, but the accepted consensus standard at this time is the ASHRAE 93-77 standard. The Department of Energy (DOE) is using the ASHRAE standard in developing its program for national certification and rating of solar collectors. Therefore, all data given in this report and future reports will conform to the ASHRAE standard.

Figure 2-7 shows many contemporary solar collectors as of the writing of this report. Data is from ASHRAE 93-77 tests. In some cases CEL has conducted the test itself. A typical CEL test report of a solar collector is given in Durlak (1979a) which is the latest report. It summarizes other reports available.

A large amount of test data on solar collectors is becoming available through the national program run by DOE, the CEL tests, and individual laboratories testing for the manufacturers. Some of this data is shown in Table 2-6, which is the basis for Figure 2-7.

Table 2-6 represents a random sampling of the many solar collectors available. It is not a comprehensive list nor is it an endorsement of any particular collector. These data were excerpted from Durlak

(1979a), Florida Solar Energy Ctr (1979), and Solar Age Magazine (1979). The main criteria for each collector in Table 2-6 is that it have an accepted ASHRAE 93-77 performance test. Other than that, collectors were chosen to provide a variety of types, materials, construction techniques, geographical locations, and cost information. A few cautions are advisable. Prices may be up to 1 year old from the publication date and should be checked if a purchase is anticipated. Manufacturers may have other models available. For example, Table 2-6 may give details for a single glazed collector and chances are the manufacturer would also have a double glazed model with valid ASHRAE 93-77 test data. The user may know of other collectors with test data available. These could be readily compared to similar models in Table 2-6.

To select a collector from Table 2-6, first note that collectors constructed of similar materials (copper, aluminum, etc.) are grouped together. Then, it is necessary to pay attention to the y intercept (called efficiency intercept in Table 2-6) which gives the highest efficiency of a collector, and the slope which gives a measure of the rate at which the collector efficiency decreases. These parameters will be used later in estimating the solar collector performance. In general the more negative the slope, the less efficient the collector. However, this must be balanced with the value of the efficiency intercept. For example, in Figure 2-7 and Table 2-6 note that double glazed collectors start out at a lower instantaneous efficiency (y-intercept) but do not lose efficiency very fast (less negative slope) so that when comparing with single glazed collectors the operating temperature (T_i) will ultimately determine which is best (see Table 2-4 also). When the cost of the collector is also considered, it becomes very difficult to "intuitively" pick a best collector in Table 2-6. The user should consider several options of collectors when using the worksheets in the later sections. In choosing a collector Figure 2-7 should be used only for qualitative judgments, while Table 2-6 should be used for typical slope and intercept values. This avoids the errors associated with trying to "read off" numbers on Figure 2-7.

Table 2-6. Solar Collector Test Results

[FRP = Fiberglass Reinforced Plastic; EPDM = Ethylene Propylene Diene Monomer (synthetic elastomer)]

Item	Model No.	Glazing Type		Absorber Coating		Efficiency Intercept (%)	Slope of Efficiency Line (Langley/°F Day)	Area of One Panel (Gross) Net (ft^2)	Cost Per Panel ($)	Remarks	Manufacturer
		Single	Double	Non-Selective	Selective						
						Aluminum Absorber Collectors					
1	3784	Glass		X		0.71	-2.07	(40.8) 37.8	-	Formed aluminum waterways	All Sunpower, Inc. Miami, FL
2	SA4		Glass		X	0.71	-1.40	(19.1) 17.4	–	Aluminum "Rollbond"	Dumont Industries Monmouth, ME
3	24-00D		FRP	X		0.61	-1.40	(26.3) 24.4	-	Aluminum "Rollbond"	General Energy Devices Clearwater, FL
4	60A	Arched Acrylic		X		0.56	-1.70	(31.0) 24.0	281	Aluminum "Rollbond"	Grumman Energy Systems Ronkonkoma, NY
5	Alum.	Glass		X		0.77	-2.27	(23.5) 22.4	282	Aluminum "Solar Bond"	Colt, Inc. Rancho Mirage, CA
						Steel Absorber Collectors					
6	711301		Glass		X	0.79	-1.02	(21.2) 21.0	252	Steel absorber and waterways	Chamberlin Manufacturing Corp. Elmhurst, IL
7	5849		FRP Tedlar Coated	X		0.61	-1.96	(11.0) 9.6	–	Steel tube and plate, microsorb coating	Solar Research Div. Brighton, MI
8	300		FRP Kalwall		X	0.52	-1.64	(19.5) 17.6	-	Copper tubes press fit to steel sheet	Sun Tank, Inc. Miami, FL
9	SCI-4		FRP	X		0.58	-1.61	(26.4) 24.7		Absorber is two welded and inflated steel sheets	State Industries, Inc. Ashland City, TN

(continued)

Table 2-6. Continued.

[FRP = Fiberglass Reinforced Plastic; EPDM = Ethylene Propylene Diene Monomer (synthetic elastomer)]

Item	Model No.	Glazing Type		Absorber Coating		Efficiency Intercept (%)	Slope of Efficiency Line (Langley/°F Day)	Area of One Panel (Gross) Net (ft^2)	Cost Per Panel ($)	Remarks	Manufacturer
		Single	Double	Non-Selective	Selective						
Copper Tubes on Aluminum Absorbers											
10[a]	SC-2000-S	Glass		X		0.63	-1.88	(19.7) 16.7	175	Copper tube on aluminum plate	Federal Prison Industries Washington, DC
11[a]	SC-2000-D		Glass	X		0.62	-1.44	(19.7) 16.7	185	Copper tube on aluminum plate	Federal Prison Industries Washington, DC
12	SC-1	FRP		X		0.63	-1.69	(32.5) 30.2	213	Copper tube on aluminum plate	Amsun Newbury Park, CA
13	Electra EC-2	Glass		X		0.69	-1.69	(22.7) 20.8	296	Copper tube in absorber of aluminum honeycomb fins	Solarnetics El Cajon, CA
14	B	FRP-Tedlar		X		0.57	-1.82	(33.1) 30.4	–	Copper tube mechanically bonded to aluminum plate	D. W. Browning Holly Hill, FL
15	31AC	Acrylic		X		0.53	-1.26	(32.4) 30.2	-	Copper tube under aluminum fins and curved relectors	Falbel Energy Systems Corp. Greenwich, CT
16	60F	Arched Acrylic		X		0.51	-1.65	(31.0) 24.0	322	Copper tube pressed between aluminum fins	Grumman Energy Systems Ronkonkoma, NY
17	CU30-WW	Glass		X		0.72	-1.63	(33.1) 29.9	336	Copper tube pressed in aluminum absorber sheet	Gulf Thermal Corp. Bradenton, FL

(continued)

Table 2-6. Continued

[FRP = Fiberglass Reinforced Plastic; EPDM = Ethylene Propylene Diene Monomer (synthetic elastomer)]

Item	Model No.	Glazing Type		Absorber Coating		Efficiency Intercept (%)	Slope of Efficiency Line (Langley/°F Day)	Area of One Panel (Gross) Net (ft^2)	Cost Per Panel ($)	Remarks	Manufacturer
		Single	Double	Non-Selective	Selective						
18	LSC-D	Glass		X		0.79	-2.27	(21.6) 20.0	350	Copper tubes imbedded in aluminum fins and plate	Lordan Solar Systems Los Angeles, CA
19	SG-18-9	Glass		X		0.63	-1.68	(20.6) 17.3	340	Aluminum fins swage fit to copper tubes	Raypak, Inc. Westlake Village, CA
20	H-1		Glass	X		0.72	-1.59	(19.0) 16.7	380	Copper tubes mechanically bonded to aluminum fins	Solar Unlimited, Inc. Huntsville, AL
21	H-2	Glass		X		0.81	-2.16	(19.0) 16.7	280	Copper tubes mechanically bonded to aluminum fins	Solar Unlimited, Inc. Huntsville, AL
All Copper Absorber Collectors											
22	C38B-H-SS	Glass			X	0.77	-1.72	(26.0) 23.5	355	Copper "Rollerbond"	Sunpower Systems Sunnyvale, CA
23	Apollo 3690	FRP		X		0.64	-1.72	(22.9) 20.7	–	Copper tube soldered to copper plate	Air Comfort Raleigh, NC
24	HW-2R	FRP		X		0.56	-1.79	(22.8) 20.9		Copper tube on copper sheet	American Solar Power Tampa, FL
25	DG-15		Glass		X	0.66	-1.10	(23.6) 21.4	–	All copper "Rollbond"	American Solar King Corp. Waco, TX
26	CO2	Glass		X		0.64	-2.19	(21.3) 18.8		Copper tube on copper sheet	Apollo Solar Energy Corp. Jacksonville, FL
27	20-34-92	Glass		X		0.59	-1.79	(22.2) 19.6	–	Copper tubes mechanically bonded to copper sheet	CBM Manufacturing Fort Lauderdale, FL

(continued)

Table 2-6. Continued

[FRP = Fiberglass Reinforced Plastic; EPDM = Ethylene Propylene Diene Monomer (synthetic elastomer)]

Item	Model No.	Glazing Type		Absorber Coating		Efficiency Intercept (%)	Slope of Efficiency Line (Langley/^{o}F Day)	Area of One Panel (Gross) Net (ft^2)	Cost Per Panel ($)	Remarks	Manufacturer
		Single	Double	Non-Selective	Selective						
28	21B	Glass			X	0.62	-1.17	(25.2) 21.1	433	Glass cover has polymer film on it; copper tube soldered to copper sheet	Daystar Corp. Burlington, MA
29	24-41C	FRP Tedlar		X		0.66	-1.79	(28.5) 26.8	--	Copper "Rollbond"	General Energy Devices Clearwater, Fl
30	–	FRP		X		0.73	-2.10	(32.0) -	350	Copper tube on copper plate	American Solar Heat Corp. Danbury, CT
31	LA100100	Glass			X	0.71	-2.01	(20.8) 18.9	355	Copper tube oven soldered to copper sheet	Sunworks New Haven, CT
32	2400-A	FRP Kalwall		X		0.67	-2.36	(24.4) 22.9	285	Copper tube soldered to copper sheet	Horizon Enterprises Homestead, Fl
33	EC-100L	Glass			X	0.73	-1.56	(19.5) 17.2	395	Copper "Rollerbond"	Novan Energy Boulder, CO
34	RCB-210	FRP Kalwall			X	0.61	-1.25	(20.6) 18.4	--	Copper tube soldered to copper sheet	Rheem Mfg. Co. Chicago, IL
35	III	FRP Kalwall		X		0.56	-2.58	(18.4) 15.4	135	Copper tube and sheet, Kennecott Terra-Light 1A	Solar Alternative, Brattleboro, VT
36	SFP-40	FRP (Filon 388)		X		0.56	-2.00	(34.3) 30.4		Copper tube press fit to copper sheet	Solar Engineering, Inc Boca Raton, Fl
37	SOL-16	FRP Kalwall		X		0.68	-2.05	(16.4) 14.7		Copper tube bonded to copper sheet	Solar One, Ltd Virginia Beach, VA

(continued)

Table 2-6. Continued.

[FRP = Fiberglass Reinforced Plastic; EPDM = Ethylene Propylene Diene Monomer (synthetic elastomer)]

Item	Model No.	Glazing Type: Single	Glazing Type: Double	Absorber Coating: Non-Selective	Absorber Coating: Selective	Efficiency Intercept (%)	Slope of Efficiency Line (Langley/°F Day)	Area of One Panel (Gross) Net (ft²)	Cost Per Panel ($)	Remarks	Manufacturer
38	1-10-D	FRP Kalwall		X		0.56	-2.01	(49.5) 42.8	~500	Copper tube soldered to copper sheet	Sun Dance, Inc. Miami Lakes, FL
39	101L		(1) Glass (1) Plexi-glass	X		0.53	1.20	(15.8) 12.9		Copper tube on copper foil	Solarec, Inc. Salem, OH
40	Solarvak	Glass			X	0.68	-0.68	(32.0)	525	Evacuated collector Copper tube under copper absorber	Solar Systems, Inc. Tyler, TX
41	CC-1F/G	FRP Filon		X		0.50	-1.40	(32.0) 23.2	260	Kennecott Terra Light copper tubes bonded to copper sheet	Vulcan Solar Industries Pawtucket, RI
42	WSD5-CR	FRP Kalwall			X	0.61	1.25	(20.0) 18.4		Copper tube soldered to copper sheet	Western Solar Development Vacaville, CA
Plastic or Rubber Absorber Collectors											
43	Sunmat	FRP		X		0.63	1.76	(32.0) 30.0	416	EPDM (rubber) absorber	Calmac Mfg. Corp. Englewood, NJ
44	1000-1	Unglazed Panel		X		0.66	5.01	(40.0) 39.4	190	Extruded polypropylene absorber with tubular sections connected by a curled web; black pigment coating	Solar Industries, Inc. Farmingdale, NJ
45	S-10	Acrylic		X		0.25	0.97	(23.1) 18.4		Copper tubing plated onto ABS plastic absorber	Solarcell Corp. Fort Lauderdale, FL
46	Solaroll TG-32		Glass	No Coating		0.71	1.66	varies	8/ft²	EPDM (rubber) absorber	Bio Energy Systems Spring Glen, NY
47	4000		Poly carbonate	X		0.70	1.98	(8.12) 8.0	7.9/ft²	Three layer extrusion of high temperature polycarbonate	Ramada Energy Tempe, AZ

continued

Table 2-6. Continued

[FRP = Fiberglass Reinforced Plastic. EPDM = Ethylene Propylene Diene Monomer (synthetic elastomer)]

Item	Model No.	Glazing Type		Absorber Coating		Efficiency Intercept (%)	Slope of Efficiency Line (Langley/°F Day)	Area of One Panel (Gross) Net (ft^2)	Cost Per Panel ($)	Remarks	Manufacturer
		Single	Double	Non-Selective	Selective						
Air Solar Collectors											
48	170A/SCF	Glass		X		0.58	-1.63	(36.0) 32.0		Aluminum absorber	BDP Co. Indianapolis, IN
49	Air		FRP	X		0.32	-0.72	(16.0) 14.4	112	Steel absorber, upper cover corrugated filon, lower cover Kalwall	Telluride Solar Works Telluride, CO
50	Mark II	FRP		X		0.47	1.20	(26.9) 31.2	375	Steel absorber	Solar Farm Industries, Stockton, KS
51	3000	Glass			X	0.59	-0.99	(30.0) 34.2	$18/ft^2$	Copper with nickel underlayment absorber plate	Solafern, Ltd Bourne, MA

[a]Same as collectors manufactured by Energy Systems, Inc., San Diego, CA.

2.1.10 Other Types of Solar Collectors. The three most common types of solar collectors are flat plate collectors, evacuated tube collectors and concentrating collectors. Due to certain cost and performance advantages, flat plate collectors have been used extensively for domestic water heating and space heating applications. Evacuated tube and concentrating collectors are used mostly in solar applications requiring very high temperatures. A brief description follows.

2.1.10.1 Evacuated-Tube Collectors: Figure 2-8 shows an evacuated-tube collector. This type of collector uses a vacuum between the absorber and the glass outer tube to significantly reduce convection and conduction heat losses.

Evacuated-tube collectors operate essentially the same as flat-plate collectors. Solar radiation passes through the outer glass tube and is abosrbed by the coated absorber. Heat energy is transferred to fluid flowing through the absorber.

Most evacuated-tube designs collect both direct and diffuse radiation efficiently, but certain types are specifically designed for more efficient collection of direct radiation. Although evacuated-tube collectors are considerably more expensive than typical flat-plate collectors, they are much more efficient when high collection temperatures are needed for operating absorption chillers or for industrial processing.

They may not be as efficient as flat-plate collectors at low-temperature applications such as domestic water heating and space heating. Comparisons can be made using data similar to that in Figure 2-7 and Table 2-6.

2.1.10.2 Concentrating Collectors: Concentrating or focusing collectors intercept direct radiation over a large area and focus it onto a very small absorber area. These collectors can provde very high temperatures more efficiently than flat-plate collectors, since the absorption surface area is much smaller. However, diffuse sky radiation cannot be focused onto the absorber. Most concentrating collectors require mechanical equipment which constantly orients the collectors towards the sun and keeps the absorber at the point of focus.

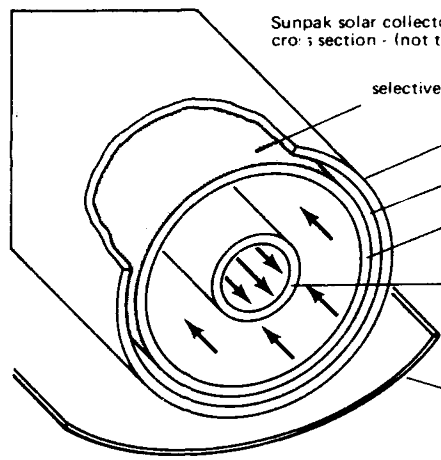


Figure 2-8. Evacuated tube solar heat collector.

There are many types of concentrating collectors. The most popular types are the parabolic trough, the linear-trough fresnel lens, and the compound parabolic mirror. Figure 2-9(a) shows a linear concentrating or parabolic trough collector. It collects energy by reflecting direct solar radiation off a large curved mirror and onto a small absorber tube which contains a flowing heat transfer liquid. The absorber tube is encased in a glass or metal tube which may or may not be evacuated. This type of collector must track the sun and can collect only direct radiation.

Figure 2-9(b) shows a linear-trough, fresnel lens collector. In this design a curved lens is used to focus incoming rays onto a small absorber plate or tube through which the heat transfer liquid is circulated. This type of collector also requires a tracking mechanism and can collect only direct radiation.

Figure 2-9(c) shows a compound parabolic mirror collector. The design of the mirrors allow the collector to collect and focus both direct and diffuse radiation without tracking the sun. Periodic changes in the tilt angle are the only adjustments necessary.

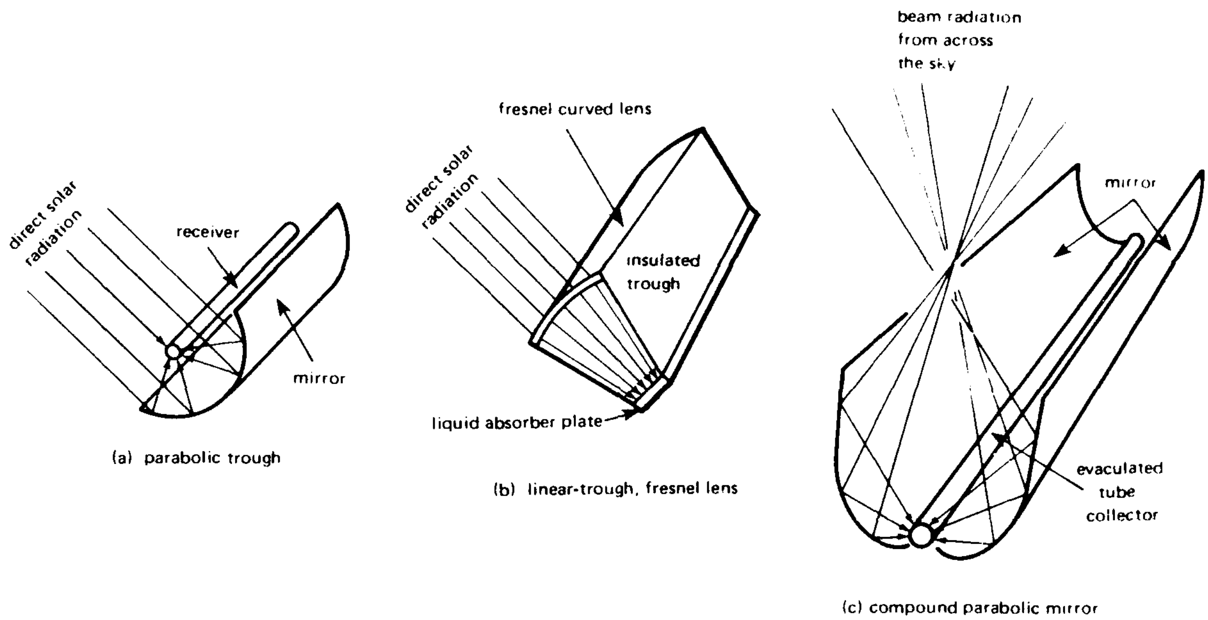


Figure 2-9. Concentrating collectors for solar energy.

Direct radiation is intercepted by only a portion of the mirror at a time, thus this collector does not collect as much solar energy as a focusing collector which tracks the sun. It is, however, less expensive to install and maintain. The absorber tube is encased within an evacuated tube to reduce heat losses.

Many other types of concentrating collectors have been developed which produce high temperatures at good efficiencies. However, the high cost of installing and maintaining tracking collectors restricts their use to solar cooling and industrial applications where extremely high fluid temperatures are required. In addition, concentrating collectors must be used only in those locations where clear-sky direct radiation is abundant.

2.2 Energy Storage and Auxiliary Heat

Since effective sunshine occurs only about 5 to 6 hours per day (in temperate latitudes), and since heating and hot water loads occur up to 24 hours a day, some type of energy storage system is needed

when using solar energy. The design of the storage tank is an integral part of the total system design. Although numerous storage materials have been proposed, the most common are water for liquid collectors and rock for air. These have the advantages of low cost, ready availability and well known thermal properties.

Precise heat storage sizing is not necessary, but economics and system design do determine the optimum range of sizes. The temperature range wherein useful heat is stored is important in determining optimum system size. If the volume of storage is too large, the temperature of the storage medium will not be high enough to provide useful heat to the building. Also, overdesigned storage requires excess floor space. If the storage is too small, the storage temperature will be too high, resulting in low collector efficiency.

Practical experience in the industry as well as computer simulations and experiments have resulted in general rules of thumb for storage sizing. These guidelines give storage sizes for which the performance and cost of active solar systems are optimized and relatively insensitive to changes within the range indicated.

The optimum size of storage for active solar systems is 15 Btu/°F/ft^2 of collector area (Kohler, 1978). The range is 10-20 Btu/°F/ft^2 (200-400 KJ/°C/m^2). For water or air systems application of the rule gives the following:

Water Systems* - Since water has a specific heat of 1 Btu/lb-°F, then 15 lb of water storage are needed per square foot of collector or considering the density of water (62.4 lb/ft^3) then 1.8 gal of storage are needed for each square foot of collector (range 1.2 to 2.0 gal/ft^2). The range in SI units is 50-100 liters/m^2.

Air Systems* - Since rock has a specific heat of 0.21 Btu/lb-°F and rock densities (170 lb/ft^3) typically contain 20%-40% voids, then the optimum storage size is 0.8 ft^3 per square foot of collector (range 0.5 to 1.15 ft^3 per square foot of collector). The range in SI units is 0.15 to 0.35 m^3/m^2.

*Storage volumes in this range will store the equivalent of overnight to one full day of heating.

In general, for equal storage capacity, the rock pebble bed would have to occupy a volume 2-1/2 to 3 times larger than a water tank. Rock storage bins have higher structural requirements, and tend to lose more heat due to their greater surface area. Rock bins generally provide good temperature stratification; contrary to practice in conventional DHW systems, stratification is desirable in both water and air solar systems. CEL has done studies to show that good stratification can add 5%-10% to overall system performances (Sharp and Loehrke, 1978). To achieve this, baffles or modified inlets to the tanks are used. To suppress convection warm water enters and leaves the top of the tank, and cold water, the bottom. In this way the hottest water goes to the load and the coldest to the collectors. A typical domestic hot water system is shown in Figure 2-10. Use of two tanks insures that when hot water from the first (tempering) tank is available, the auxiliary heat will not come on; also less total fuel will be used to bring the smaller second tank up to temperature. Single tank arrangements while possible and economical are not recommended due to the fact that they tend to activate the heating element every time there is a draw of water rather than wait for the solar collectors to provide additional heated water. The two-tank arrangement avoids this control problem. Two-tank arrangements are suited to retrofits since the second tank (the water heater) is already there. A variation would be to use a heat exchanger (copper coil) in the tempering tank collector loop for freeze protection. The tempering tank could then be an inexpensive unpressurized tank.

Another method of heat storage in air systems that is currently being investigated is latent heat storage. Latent heat is stored in a material as it changes phase from a solid to a liquid. Materials which have melting points near the temperatures supplied by solar collectors store heat as they melt and release it as they resolidify. The two materials which have received the most attention are salt hydrates and paraffins.

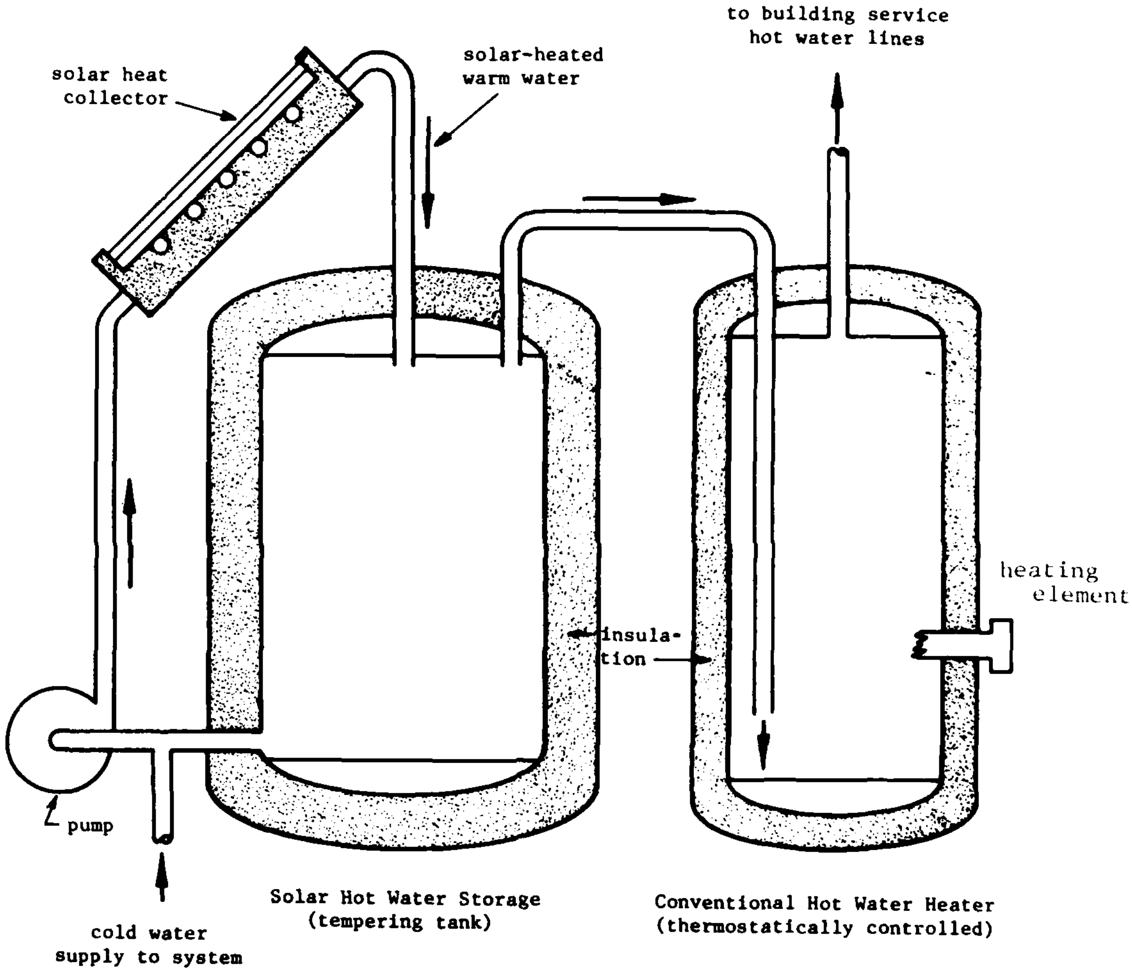


Figure 2-10. Schematic of potable hot water heating system using solar storage (tempering) tank ahead of conventional fueled or electric service water heater.

The advantage of latent heat storage is that it can store very large quantities of heat per pound of storage material. Therefore, less volume should be required for latent heat storage than for heat storage in rock beds. However, problems of slow solidification and low heat conductivity retards effective heat transfer to and from the material. As a result, a large surface area-to-volume ratio is required, which significantly increases the effective volume of latent storage.

Storage materials used are often expensive when compared to rock. In addition, they must be packaged in individual containers to allow adequate heat transfer area. Many latent heat materials cannot withstand frequent recycling and must be replaced periodically. Research is being done by CEL and others to develop practical latent heat materials which can withstand extended recycling. CEL is investigating a dissolved salt storage unit that uses immiscible liquids for the heat exchange surface which greatly reduces the problem of crystallization during recycling. Initial tests have been encouraging.

Another major drawback of latent heat storage is that heat is stored at an average temperature with essentially no thermal stratification occurring in the storage unit. A high level of thermal stratification maximizes thermal performance because low temperature fluid can be delivered to the collectors and high temperature fluid can be delivered to the heat load. For example, the high degree of thermal stratification in rock-beds results in the delivery of 79°F air to the collector and 120°F to 150°F air to the heat load. In comparison, latent heat storage in Glauber's salt occurs near an average temperature of 90°F; thus air at 90°F is delivered to both the collectors and the heat load.

Due to the problems discussed, latent heat storage has not received widespread use.

Since it is not economically justifiable to store huge quantities of heat, most solar systems cannot be depended on to provide 100% of the buildings needs. Depending on the geographical area and size of the system, about 40% to 80% of the heat requirement is the average to design for. Therefore auxiliary heaters are necessary. They should

be sized to provide all the energy requirements, although in some cases, again depending on location, it may be possible to increase storage volume and provide less than 100% backup auxiliary heat. This is especially true if the use of passive solar designs can be incorporated with active systems. Passive designs are discussed briefly in Section 2.6.

The auxiliary heater should operate automatically as needed, use the most economical fuel, and share a common heat delivery system with the solar system. Often a heat pump is a good choice in that it can serve both as an auxiliary heater and work together with the solar system. In retrofit situations, the existing heater would be the choice.

2.2.1 Storage Tanks. Water may be stored in a variety of containers usually made of steel, concrete, plastics, fiberglass, or other suitable materials.

Steel tanks are commercially available and have been used for water storage. They are available in many sizes and are relatively easy to install. However, steel tanks are susceptible to corrosion and should be lined or galvanized. Dissimilar metal at pipe connections should be separated by high temperature rubber connections or galvanic corrosion will occur. Steel tanks must be well insulated to minimize heat losses.

Concrete tanks are durable, but may be difficult to install. Concrete tanks cast in place, prefabricated septic tanks, or large diameter pipes may be used for water storage. A high temperature sealant or lining should be applied to the interior of the tank to prevent seepage of water through the tank. Although concrete is less conductive than steel, concrete tanks should also be insulated to reduce thermal losses. Leaks are difficult to repair.

Fiberglass and plastic tanks are corrosion resistant and easily installed. They are available in many shapes and sizes. Although many commonly fabricated tanks will begin to soften at temperatures above 140°-160°F, there are more expensive, specially fabricated tanks available that can withstand temperatures up to 250°F. The types of plastics needed to store large quantities of water at high temperatures can be more expensive than steel.

When storage tanks are to be custom made, a calculation of heat loss against expected fuel cost inflation will almost always justify increasing insulation around the tank to R-19, 6 inches, compared with the usual 2 inches. HUD Intermediate Minimum Property Standards (U.S. Dept HUD, 1977) requires that tank losses be limited to 10% in 24 hours. Usually R-19 insulation will satisfy this requirement.

Costs of storage tanks vary considerably depending on the quality of construction and the distributor. Table 2-7 summarizes advantages and disadvantages and Table 2-8 gives approximate comparative costs for tanks of various materials. All storage tanks for liquids should be located so that if they leak, damage to the building will be prevented. The cost of housing the tank or burying it must be included in the total cost of the solar heating system. Buried tanks must be protected from ground water, and buoyant forces resisted. Tank must be reasonably accessible for repairs. In very mild or warm climates, outdoor location may be feasible.

2.3 Domestic Hot Water Systems (DHW)

Domestic hot water systems (without space heating) may use lined, insulated, pressurized tanks similar to the conventional water heater. Appropriate temperature and pressure relief valves must be used. Since it is possible for solar collectors to reach very hot temperatures, a tempering or mixing valve should be used. A typical two-tank installation with proper valves and connections would be as shown in Figure 2-11 (Cole et al., 1979).

To size the collectors and storage tank it is necessary to estimate or measure the hot water consumption of the facility or building. For typical family residences, 20 gal/day/person of hot water is normally consumed. If it is estimated the hot water consumption is larger than average, use 30 gal/day/person. So, 80 to 120 gal/day should serve a typical four-person family. For estimates of other facilities refer to Table 2-9, which gives water consumption for many kinds of facilities (Werden and Spielvogel, 1969). Although these are for conventional facilities, they should provide a reasonable estimate of Navy facilities (barracks, etc.) for lack of any better data.

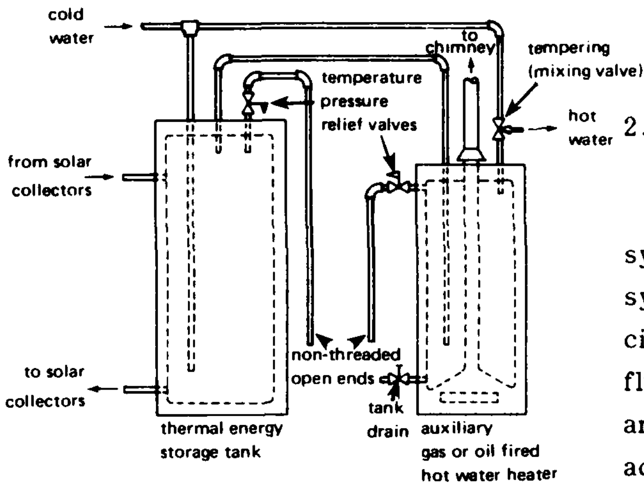


Figure 2-11. Typical DHW installation.

2.4 Thermosyphon Systems

A variation of the DHW system is the thermosyphon system which uses the principle of natural convection of fluid between a collector and an elevated storage tank. Its advantage is that no pump or controller are needed. The bottom of the tank should be mounted about 2 feet higher than the highest point of the collector. This is the main disadvantage in that structural requirements will often prohibit the weight of a water tank on a high point of the structure. Also, since the thermosyphon system is connected directly to the potable water supply it cannot be protected from freezing. A heat exchanger cannot be effectively used in this system.

Thermosyphon systems have operated successfully for many years in the Middle East and Australia. CEL has installed a thermosyphon system and experimental data collected to date has indicated that overall system performance is only slightly less than a system that uses a pump. This type of system can be considered for nonfreezing climates where structural requirements permit.

2.5 Space Heating and DHW Systems

Space heating systems are a simple extension of the domestic hot water (DHW) systems. The collectors and storage tank need to be resized to provide the greater loads. A heat delivery system is added and the auxiliary heater (or existing heater) is connected in as backup. The design of the space heating system, if a retrofit, will depend on the existing system. Water-to-air heat exchangers may be placed in existing ductwork, in which case, an unpressurized, unlined tank may be used and represents a minimum heating system as in Figure 2-12.

Table 2-7. Advantages and Disadvantages of Tank Types (Cole et al., 1979)

Advantages			
Steel Tank	**Fiberglass Tank**	**Concrete Tank**	**Wooden Tank With Liner**
Cost is moderate.	Factory-insulated tanks are available.	Cost is low.	Cost is moderate.
Steel tanks can be designed to withstand pressure.	Considerable field experience is available.	Concrete tanks may be cast in place or may be precast.	Indoor installation is easy.
Much field experience is available.	Some tanks are designed specifically for solar energy storage.		
Connections to plumbing are easy.	Fiberglass does not rust or corrode.		
Some steel tanks are designed specifically for solar energy storage.			
Disadvantages			
Steel Tank	**Fiberglass Tank**	**Concrete Tank**	**Wooden Tank With Liner**
Complete tanks are difficult to install indoors.	Maximum temperature is limited even with special resins.	Careful design is required to avoid cracks and leaks.	Maximum temperature is limited.

(continued)

Table 2-7. Continued

Disadvantages (continued)			
Steel Tank	Fiberglass Tank	Concrete Tank	Wooden Tank With Liner
Steel tanks are subject to rust and corrosion.	Fiberglass tanks are relatively expensive. Complete tanks are difficult to install indoors. Fiberglass tanks must not be pressurized.	Concrete tanks must not be pressurized. Connections to plumbing are difficult to make leaktight.	Wooden tanks must not be pressurized. Wooden tanks are not suitable for underground installation.

Table 2-8. Storage Tank Costs[a]

Type of Installations	Cost Per Gallon for Tank Size (gal) -						
	80	120	300	500	1000	2000	4000
Steel, unlined, nonpressurized[b]			1.10	0.70	0.65	0.60	0.50
Steel, unlined, 125 psi[b]	2.75	2.00	1.45	1.30	1.25	1.20	1.15
Steel, glass lined,[b] 125 psi	3.70	3.80	3.20	-	-	-	-
1/8-inch fiberglass liner for unlined tank				1.05	0.95	0.70	0.55
Steel, stone lined, 125 psi	4.80	4.10	-				
Fiberglass tank, or polyethylene, nonpressurized	2.00	2.00	1.60	1.50	1.40	1.20	1.00
Concrete 6-inch insulation and sheath	1.50	1.40	1.20	1.10 1.00	1.00 0.80	0.80 0.60	0.70 0.50
Normal installation above ground including pad	1.00	1.00	1.00	0.95	0.85	0.60	0.40

[a]All prices, \$/gal, Dec 1978, Los Angeles area.

[b]Includes supports and fittings; add \$0.15/gal for phenolic lining of unlined tanks.

Table 2-9. Hot Water Demands and Use for Various Types of Buildings

Type of Building	Average Day
Men's Dormitories	13.1 gal/student
Women's Dormitories	12.3 gal/student
Motels (number of units):	
20 or Less	20.0 gal/unit
60	14.0 gal/unit
100 or More	10.0 gal/unit
Nursing Homes	18.4 gal/bed
Office Buildings	1.0 gal/person
Food Service Establishments	
Type A - Full meal restaurants & cafeterias	2.4 gal/average meals/day[a]
Type B - Drive-ins, grilles, luncheonettes, sandwich & snack shops	0.7 gal/average meals/day[a]
Apartment Houses (number of apartments):	
20 or Less	42.0 gal/apartment
50	40.0 gal/apartment
75	38.0 gal/apartment
100	37.0 gal/apartment
Over 130	35.0 gal/apartment
Elementary Schools	0.6 gal/student[a]
Junior & Senior High Schools	1.8 gal/student[a]

[a]Per day of operation.

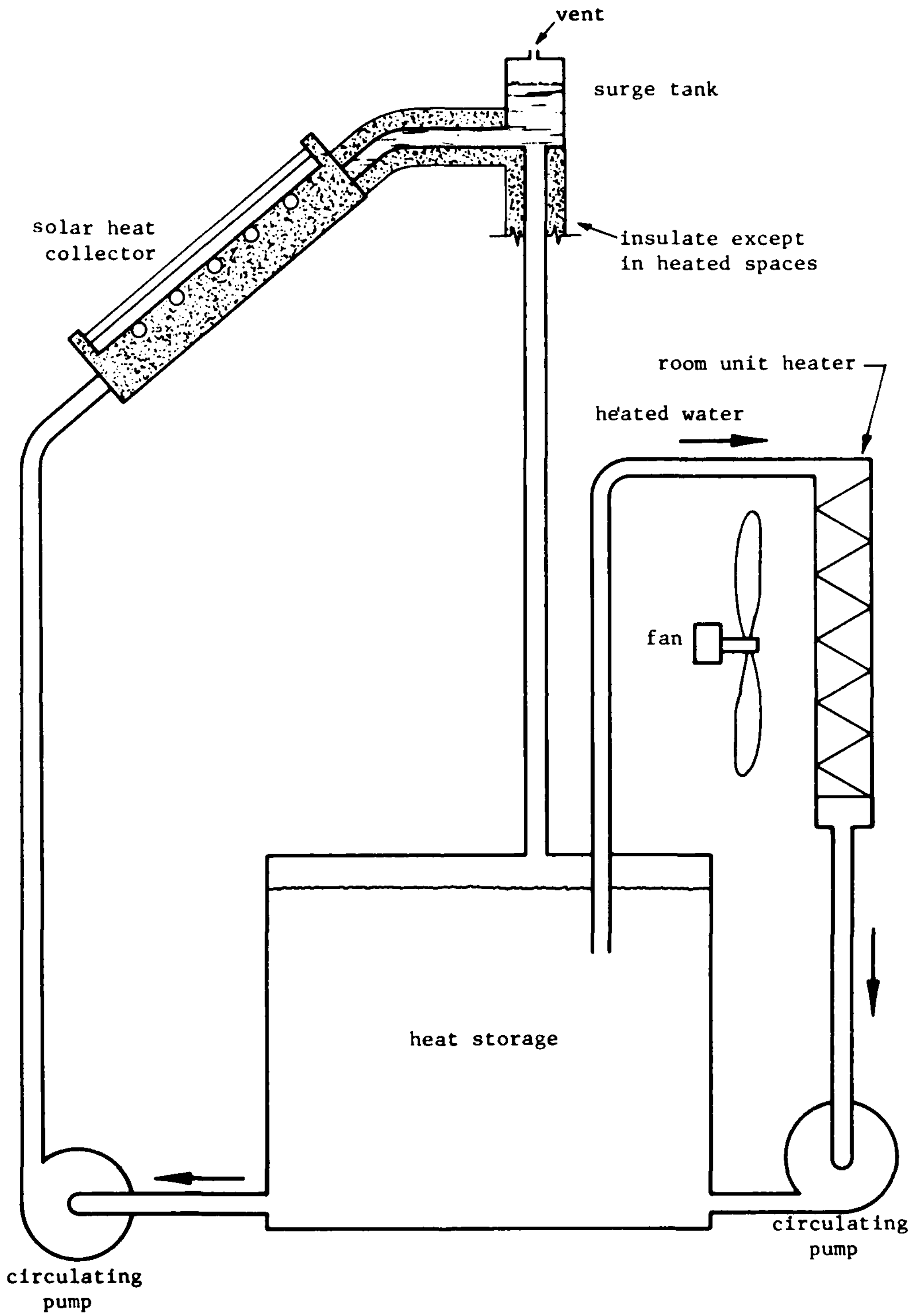


Figure 2-12. Minimum heating system, showing relationship of collector, storage, and room unit heater.

To provide corrosion and/or freeze protection the use of a closed collector loop and heat exchanger are required as in Figure 2-13. The cautions given in Figure 2-5 regarding toxic heat transfer fluids must be observed.

The most versatile system would be as shown in Figure 2-14 in which space heating and DHW are provided. Remember that 100% back-up capability is needed but that oversizing is not necessary. Auxiliary heat can be used directly such as a heat pump or separate furnace or it could be added to the main storage tank using a heat pump, a separate boiler, or electrical resistance heating.

Domestic hot water could be added to Figure 2-12 and 2-13 by adding a preheat coil in the storage tank. Figure 2-12 has the potential to provide some building cooling by using the collector at night to radiate heat to the sky and storing cool water for use during the day. Or a heat pump could be used to cool the building, reject heat to the storage tank during the day, and then, as before, cool the tank at night through the solar collectors. Unglazed collectors are superior to glazed collectors for this application. There are many variations that could be used with the configurations given in Figures 2-12 through 2-14.

Air type space heating systems are receiving increased attention and a typical system is shown in Figure 2-15 (see Table 2-1 for advantages of air versus liquid). The heat storage tank is replaced by a rock bed (nominally 1-3 inch diameter). Rock provides very desirable temperature stratification. Designs should emphasize minimum pressure drop through the rock bed. The rocks can be stored in a bin, which should be insulated, or beneath the building if this is feasible. Heat collected by the collectors is blown through the rock bed from top to bottom. Heat is delivered from storage to the building by circulating air in the reverse direction, bottom to top. Note that in contrast to water storage, heat cannot be added to and removed from the rocks at the same time.

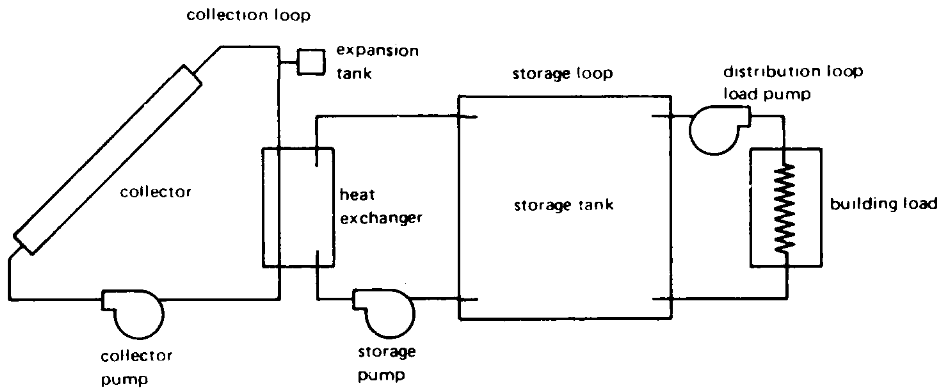


Figure 2-13. Space heating system with closed collector loop.

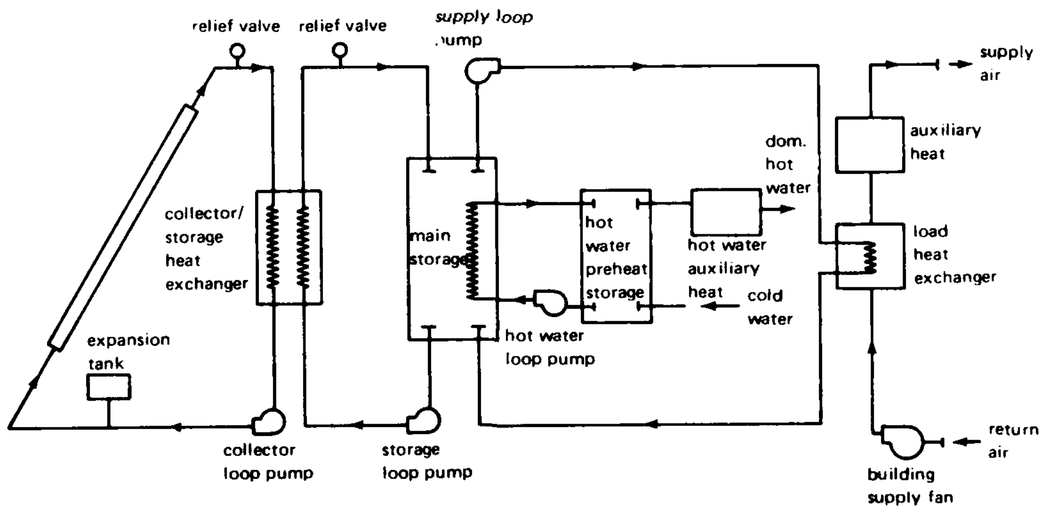


Figure 2-14. Space heating and domestic hot water system.

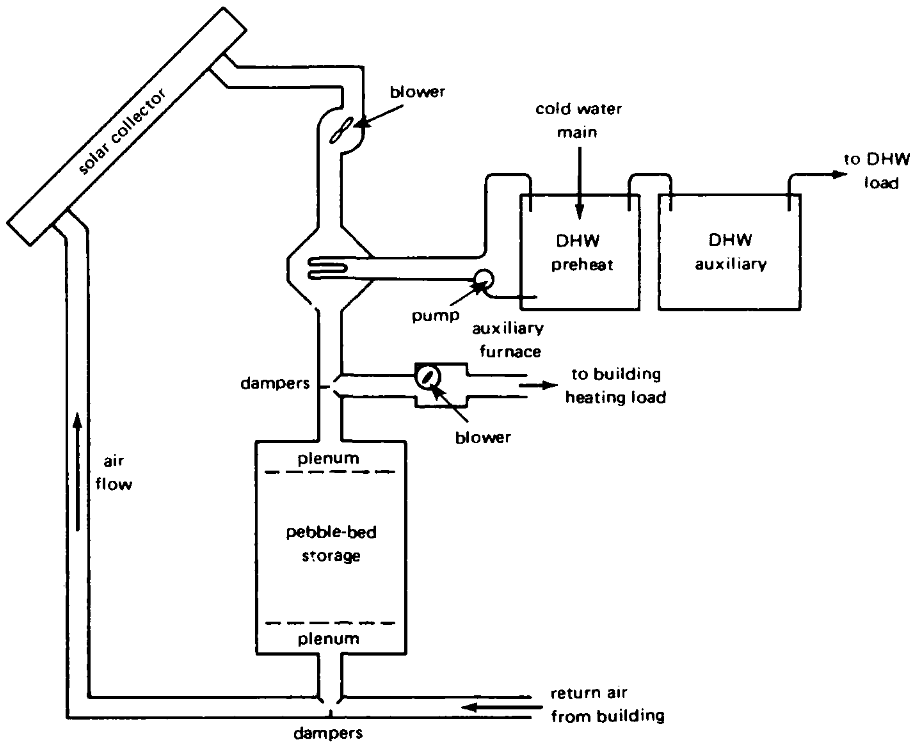


Figure 2-15. Typical air-type space heating system.

During heat collection, the rocks at the top of the bin will attain a temperature nearly equal to that of the incoming solar heated air, while the air leaving the storage will be delivered to the collectors at the minimum temperature of the rocks. The conduction between the rocks is small, thus with no air circulation the rock bed will remain stratified with the top of the rock bed warmer than the bottom. Also, limited conduction and convection in the rock-bed significantly reduces heat loss from the rock-bed.

Heat is drawn from the storage by circulating building air directly through the rock bed from bottom to top. The air will then be delivered to the building at a temperature near the maximum temperature of the collectors. If additional heat is required, supplementary heat is added down-stream from the storage unit. This system allows the rock bed to deliver useful heat until all of the rocks are at room temperature. See Duffie and Beckman (1974) for design of packed bed rock storage.

A variation is a no-storage air heating system which circulates heated air when available. Performance is limited to daytime heating due to the lack of storage, but such systems are well suited to warehouses and factories which have mainly daytime operations.

Domestic hot water is provided in Figure 2-15 by pumping the water in the preheat tank through an air-to-water heat exchanger placed in the return air duct from the collectors. This is not very efficient and is one of the disadvantages of the air system. It would, perhaps, be just as cost effective to have a separate small liquid system (say two collectors) to provide domestic hot water.

Both air and liquid space heating systems require a heat delivery network to transfer heat from storage to the building. Most of the buildings in the United States are heated by circulation of warm air through the building. The air is usually heated in a central location and ducted to the individual rooms. This method is used particularly in residential buildings.

Hydronic heating is another common heat distribution method. In hydronic heating systems hot water or steam is circulated through pipes to "convectors" located in the individual rooms of a building. Modern hot water convectors are comprised of one or more finned tubes located on the wall near the floor. These baseboard heaters deliver heat to the room mainly by convection as air moves through the fins.

A less common heating system consists of lengths of tubing embedded in the floors, walls, or ceilings of the living space. Warm water is supplied to the tubes by a boiler and the heat is transferred to the room by convection and radiation.

2.5.1 Heat Distribution for Liquid-Type Solar Systems. The temperature requirements of a hydronic heating system are dependent on the amount of heat exchanger surface. Most baseboard heaters have comparatively small surface areas, so they require higher temperatures, typically about 180°F. If larger heat transfer areas are available as in older or modified hot water systems, temperatures of 120°F may be sufficient. Temperatures of 100°F to 120°F are adequate for the system which uses entire floors, walls, and ceilings as radiator surfaces (Lof, 1977).

During the winter, typical liquid-type solar systems are seldom operated at delivery temperatures above 150°F. Thus it is evident that the use of solar heated water in standard baseboard heaters is impractical. Only modified baseboard heaters of adequate size or radiant panels are suitable for use in hydronic systems which use solar heated water.

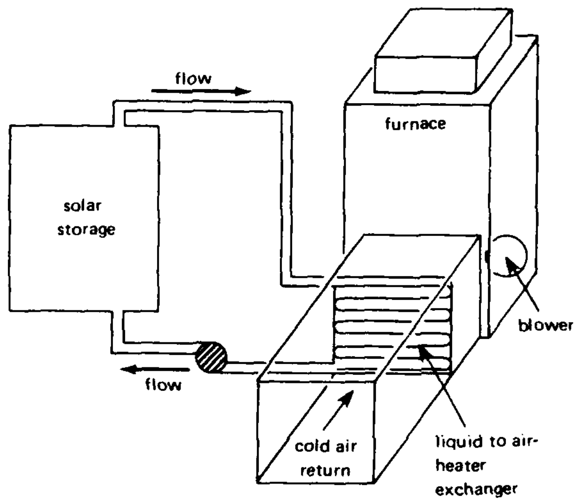


Figure 2-16. A liquid-to-air heat delivery system.

One of the most economical means of auxiliary heat supply and heat distribution for liquid-type solar systems involves the use of a warm air system. A typical system is illustrated in Figure 2-16. In this system the warm air furnace is located downstream from a liquid-to-air heat exchanger which is supplied with solar-heated water. The furnace can then serve to boost air temperature when insufficient heat is available from the solar heated water, or it can meet the full heat load if no heat is available in solar storage. Auxiliary heat can be supplied by a gas, oil, or electric furnace, or by the condenser of an air-to-air heat pump.

Another method of heat distribution involves the use of a water-to-air heat pump which draws heat from the solar storage tank and pumps it to a condenser coil which is placed in a central air duct. The advantage of this system is that it can effectively use heat from solar storage at temperatures down to 45°F, thus more of the stored heat is available. Also, average storage temperatures are lower, resulting in significantly increased collector efficiency. Heat pumps are discussed in Section 2.5.3.

2.5.2 Heat Distribution for Air-Type Solar Systems. The pipes and pumps of the liquid-type system are replaced by air ducts and fans. The warm air system is obviously the best heat distribution system for use with an air-type solar system. The ability to circulate building air directly through the collectors is one of the major advantages of an air-type solar system. The rock bed storage also works best with a warm air system.

Although warm air as low as 100°F can be used to heat an occupied building, most existing warm air systems are sized assuming warm air temperatures of 120°F to 150°F. Typical mid-day collection temperatures usually range from 130°F to 170°F. Maximum storage temperatures are typically around 140°F at the end of the collection period. Thus the heating load can be met by the temperature of the solar heated air a large portion of the day. When storage temperatures are insufficient to maintain the desired temperature in the building, heat from an auxiliary source must be added to supplement the solar heated air. The auxiliary furnace is located downstream from the rock bed so that the rock bed serves as a preheater for the furnace. This arrangement allows the rock bed to deliver useful heat until all of the rocks are at room temperature.

An air handler unit provides the dampers and blowers necessary to direct air circulation between the solar collectors, rock-bed, and building as needed. An air handler unit may be more expensive than the combined cost of individual dampers and blowers, but it will probably be less expensive to install. It is also more compact.

2.5.3 Heat Pumps. Heat pumps have been mentioned in previous sections as a possible choice for auxiliary heaters. Some manufacturers are combining solar systems with heat pumps for the purpose of reducing auxiliary energy costs. When a heat pump and a solar system are combined in this manner, the system is usually called a solar assisted or solar augmented heat pump (SAHP) system.

Solar assisted heat pump systems can be configured in many different ways. For example, the solar collectors can be either water or air types, the heat storage medium can be water or a solid material

such as rock or brick, and the heat pump can be of either the air-to-air design or the water-to-air design. But heat pumps have a characteristic which can limit their effectiveness: the efficiency and capacity of a heat pump decreases as the temperature of the heat source (usually outdoor air) decreases. This deficiency can be overcome, however, by using solar collectors to gather the sun's energy for the purpose of keeping the heat source in the temperature range required for efficient heat pump operation.

Air-to-Air Heat Pumps - The air-to-air heat pump functions very well as an auxiliary heater at temperatures down to 20°F. Below these temperatures, it suffers in efficiency and performance. When solar assisted by heat from a rock-pebble storage bed and air collectors, the heat pump adds much to the performance of the solar energy system.

Without such a solar assist, air-to-air heat pumps have limited utility in cold climates. Their use should be carefully checked with the local utility and pump manufacturer.

The heat pump also provides cooling during the summer. It thus has year-round utility. Heat pumps should be comparison-shopped. The purchaser should look at the cost, performance, service, and expected life. Units differ considerably from manufacturer to manufacturer. (Montgomery, 1978).

Liquid-to-Air Heat Pumps - The liquid-to-air heat pump is an ideal auxiliary heater when coupled with liquid solar storage. It operates at very low cost. And it greatly enhances solar energy collection by drawing down the temperature of the solar storage water to as low as 45°F. It should be considered for all installations, except those with existing fossil fuel furnaces and no need for summer cooling. (Montgomery, 1978).

Out of the many SAHP configurations which could be used, the two most in use are called the "series" and "parallel" configurations. Figure 2-17 is a series SAHP system. When the system is used for heating, water from the storage tank is circulated through water cooled collectors where it is heated before returning to the storage tank. Warm water from the storage tank is also circulated through a water-to-air heat pump. Heat is removed from the water and transported to the indoor air by the heat pump and the water returns to the storage tank at a lower temperature. If heat is added to the water in the tank faster than it is removed by the heat pump, the temperature of the water will rise. When the water temperature is high enough (about 104°F), heat can be extracted directly from the water by means of water-to-air heat exchanger. In this mode of operation, the heat pump is shut off. Auxiliary electrical resistance heaters are provided to make up the balance of the heat load if the heat from the heat pump or water-air heat exchanger is not sufficient to meet the demand.

When used for cooling, the heat pump transports heat from the building to the water in the storage tank thereby causing the temperature of the water in the tank to rise. During spring and fall, when it is not unusual to have a light cooling load during the day and a light heating load at night, the heat in the storage system is simply shuttled from the building to storage during the day and from storage to the building at night, and the solar collectors are used only to make up for lost heat. During periods of prolonged cooling demand, the heat pumped into the storage tank might be sufficient to cause the temperature of the water to rise to where the heat pump will no longer operate. Thus, provision must be made for rejecting excess heat. One method is to add a cooling tower to the system to cool the water. Another method is to circulate water through the solar collectors at night and reject heat by radiation to the night sky. During periods of high cooling load it is not desirable to also add heat to the storage tank by circulating water through the solar collectors. Therefore, when the system is in the cooling mode the solar collector circuit can be used to heat domestic hot water.

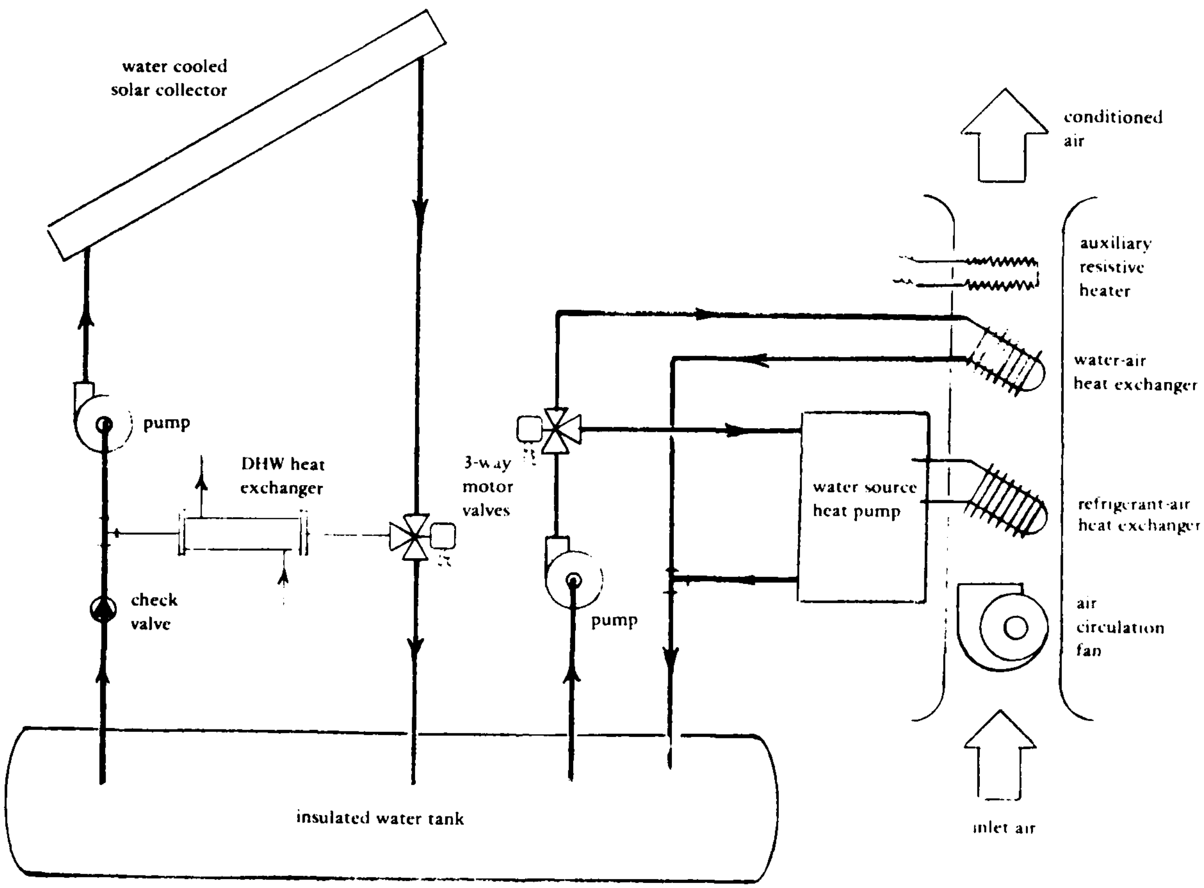


Figure 2-17. Series-connected, solar-assisted heat pump system.

The "parallel" SAHP system is shown in Figure 2-18. The solar heating system and the heat pump operate in parallel. Solar heat is used directly rather than being transferred to a storage medium and then transported into the building with a heat pump. This system is essentially a direct solar heating system with an air-to-air heat pump as a backup heating system.

The choice of a "best" system is difficult to make due to the many variables involved. For example, in addition to the two configurations shown in Figures 2-17 and 2-18, one could examine a series system with low cost (unglazed) collectors, or a series system with air-collectors and rock storage, or a parallel system with low cost collectors, etc. Each system would be highly dependent on geographical location, type of construction, etc. One such analysis done at CEL comparing several systems to a stand-alone air source heat pump, showed the "parallel" system to have the best comparative performance (Kirts, 1978). More information about heat pump systems can be found in Kirts (1978).

Each heat pump configuration should be considered on a case-by-case basis. The analysis of these systems is beyond the scope of the worksheets given in this report, and the reader is directed to more sophisticated computer programs such as those in Durlak (1979b).

2.6 Passive Systems

A "passive" solar energy system is one which uses the building structure as a collector, storage and transfer mechanism with a minimum amount of mechanical equipment. Some would include a thermosyphon system in this definition. As a rule, these systems are generally difficult to retrofit. Another disadvantage is that the owner or occupant may be required to perform daily tasks, such as covering a south facing window at night, opening and closing shutters, etc. This is particularly significant in Navy housing where the occupants are more transient and have less incentive to do these maintenance items. Although the specific arrangements vary, all of these systems rely on direct solar heating of storage. The storage then heats the house. A few examples are shown in Figure 2-19 (Barnaby et al., 1977).

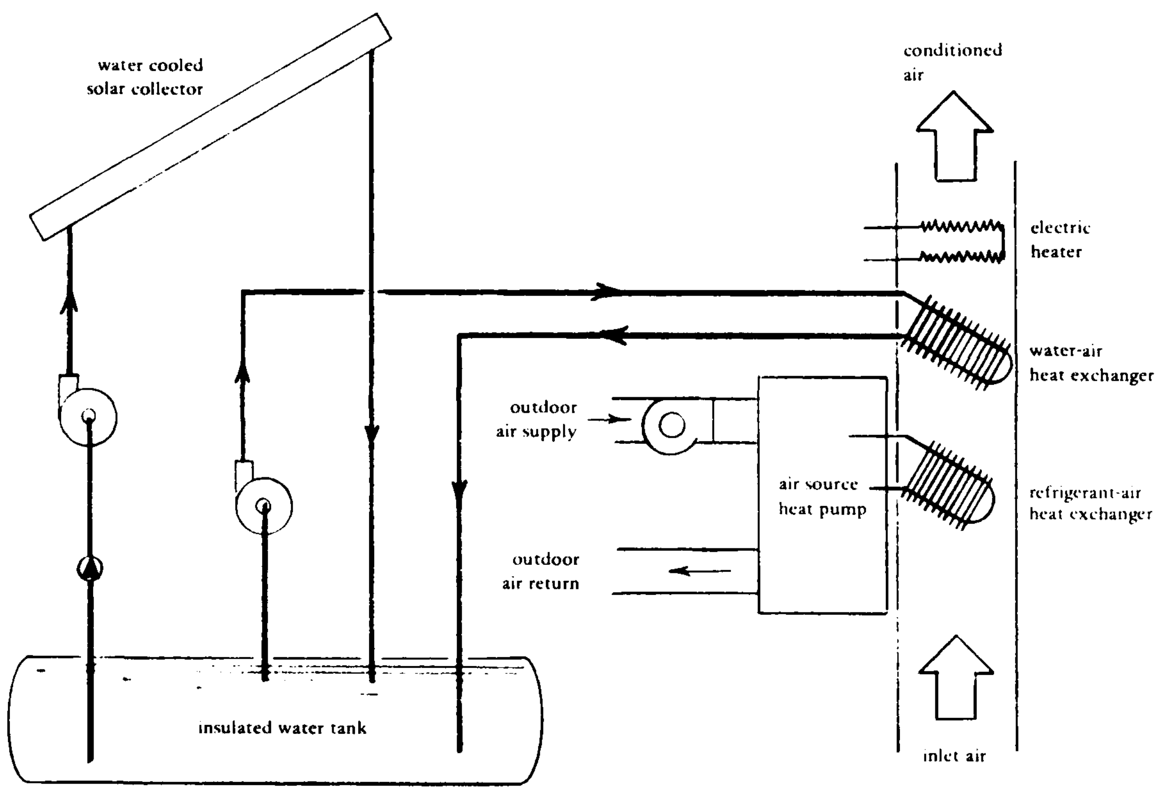


Figure 2-18. Parallel-connected, solar-assisted heat pump system.

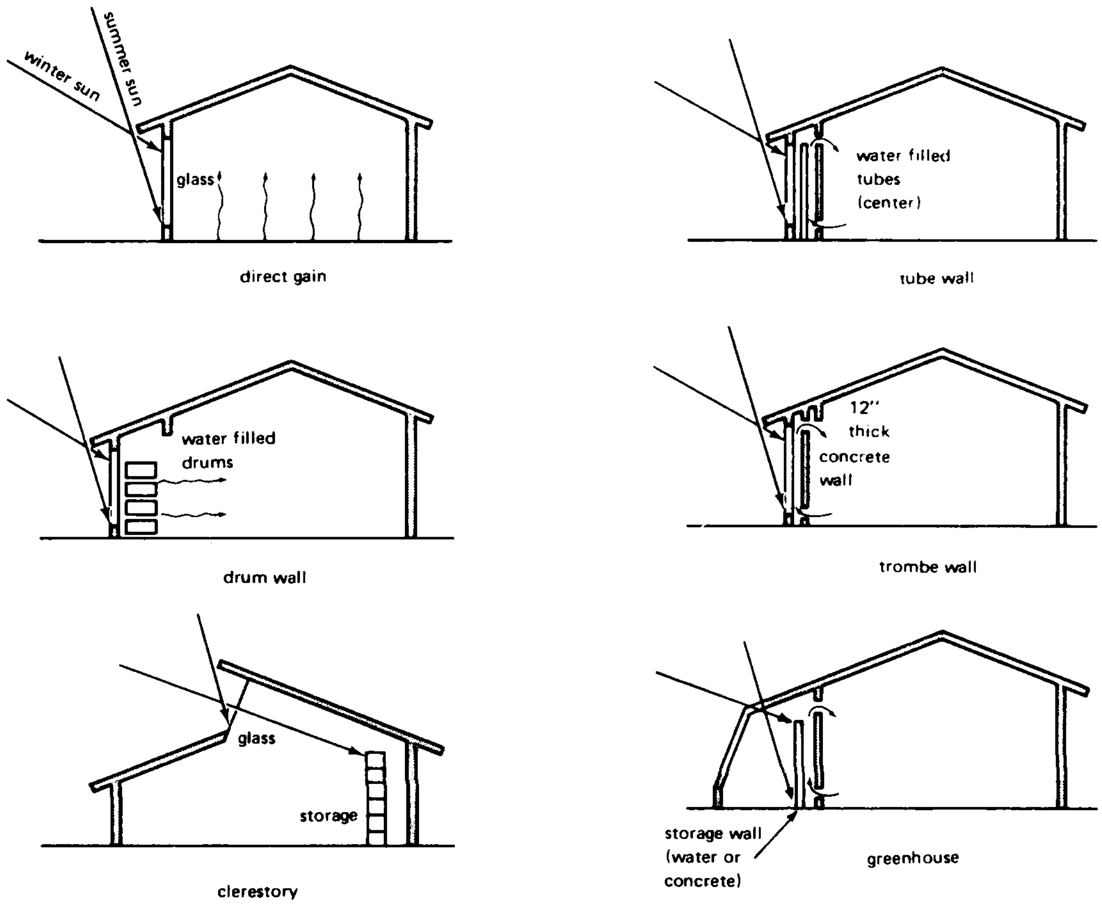


Figure 2-19. Passive solar energy systems.

Given the solar gain available on a vertical surface, the simplest and most obvious means of solar heating is just to let the sun shine in through large, south-facing windows. In fact, in a house with any south-facing windows, that's what's already happening to some degree. But the sunshine through the windows seldom heats the whole house. There are two reasons for this. First, most houses do not have enough south-facing glass. Second, houses lack enough storage to soak up the heat and keep it until night. Even rooms that overheat during the day cool off all too rapidly in the evening.

On many buildings it's possible to add south-facing windows or skylights to increase direct solar heating. However, the extra window area can cause a "fry or freeze" situation unless storage is added as

well. There must also be provisions for getting heat from the rooms receiving sunlight to the rest of the house. Providing such storage and delivery of solar heat gained through windows is the basis of passive solar heating systems.

As shown in Figure 2-19 the type of storage used and where it's located with respect to the windows varies for different passive systems. Tall metal tubes can be used to hold water instead of drums. Entire walls of solid concrete or grout-filled masonry store solar heat well. Slab floors can absorb solar heat coming in through windows, skylights, or greenhouse glass.

In each of these systems, the sunlight coming in through the glass must shine directly on the storage. If it doesn't, the storage can't absorb enough solar heat to provide much warmth for the house. Most passive systems deliver heat to the rest of the house "naturally" - that is, the heat moves by itself without use of pumps or fans. There is some natural regulation of how fast heat moves from the storage into the house - the colder the house gets, the faster the heat is drawn out of the storage. That's how the drum wall works. In other passive systems, solar heat is "trapped" between the glass and storage (in the air space between the glass and a concrete wall, or in an entire greenhouse), and the amount of heat allowed into the house is controlled by opening and closing vents, either manually or automatically.

The performance of passive systems depends not just on how much solar heat they can collect, but also on how much of that heat is lost through the glass at night. The most common solution to the problem of heat loss is to install movable insulation (such as insulating curtains) between the glass and the storage. The curtains or other devices are moved during the day to let the sunshine in, and closed at night to reduce heat loss. Certain conditions must be present to do a simple passive retrofit. Since the basis for passive heating is to "let the sun shine in," the building must have extensive south-facing windows or skylights or places where they can be added. In addition, there must be a place close to the windows where storage can be located. The storage must receive midday sun. The problem here is that drums of water and masonry walls are so heavy that most existing floors can't

support them. If the floor is not strong enough, there are at least two possible alternatives. One is to put the water or masonry wall on its own foundation on the exterior of the south wall. Another is the technique of turning a room addition into a solar heater that provides warmth for the rest of the house as well.

As with active solar systems and heat pumps, there are endless variations of the passive technique, limited only by one's imagination. There are systems that use water on the roof (SKYTHERM of Harold Hay) to absorb heat directly, and there are clever ways to insulate glass at night by blowing styrofoam beads between two glass panes (BEADWALL of Steve Baer). Also natural objects such as earth berms to protect from winds and trees which shade in summer and let light pass in winter should be considered. Figures 2-20 through 2-23 show various representations of some of these passive techniques used either by itself or in conjunction with air collectors and thermosyphon systems.

Although passive systems are rather simple in construction and design, their performance analysis is often complicated by a vast interplay of many components. Mazria (1979) is a good source of design information for passive systems. CEL plans to do some work in this area in the future. In the meantime, there are some "rules of thumb" that should be useful for passive designs:

(1) South-facing passive storage walls in direct sunlight should have a <u>minimum</u> of 30-lb water storage or 150-lb masonry (concrete) storage per square foot of south vertical glazing. If the storage media is not located in direct sunlight, four times this amount will be needed (Balcomb, undated). Mazria (1979) recommends at least 5-6 gallons water storage (about 45 lb) per square foot of south glass.

(2) Shading of south windows should be used to reduce summer and fall overheating. One effective geometry is a roof overhang which will just shade the top of the window at noon (solar time) sun elevation of 45° and will fully shade the window at noon sun elevation of 78°F (Balcomb, undated).

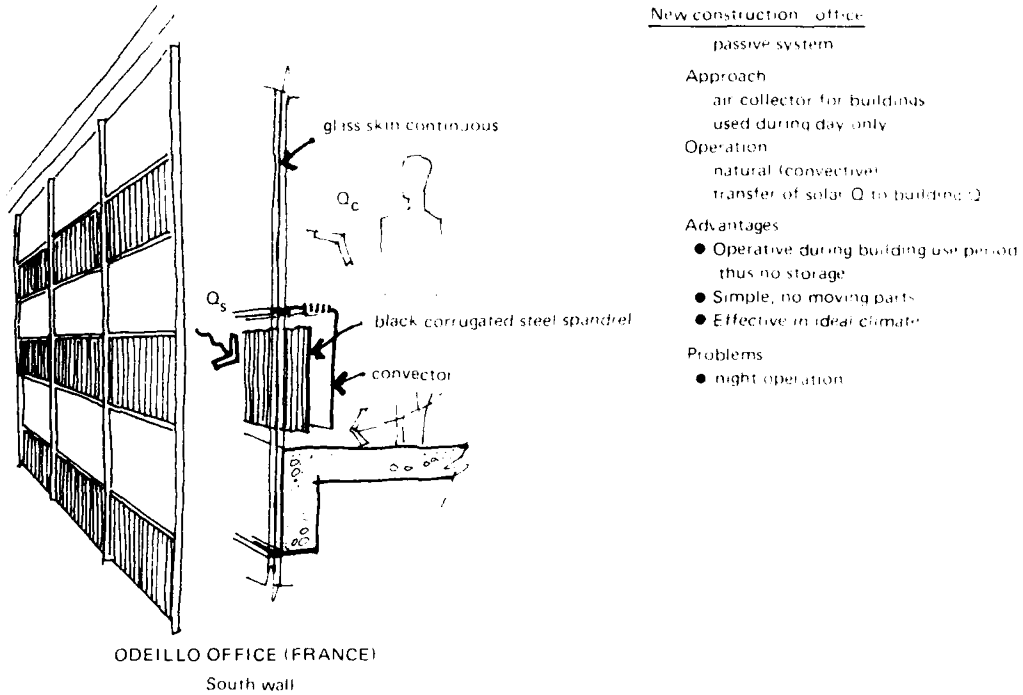


Figure 2-20. New-construction (office) passive solar energy system.

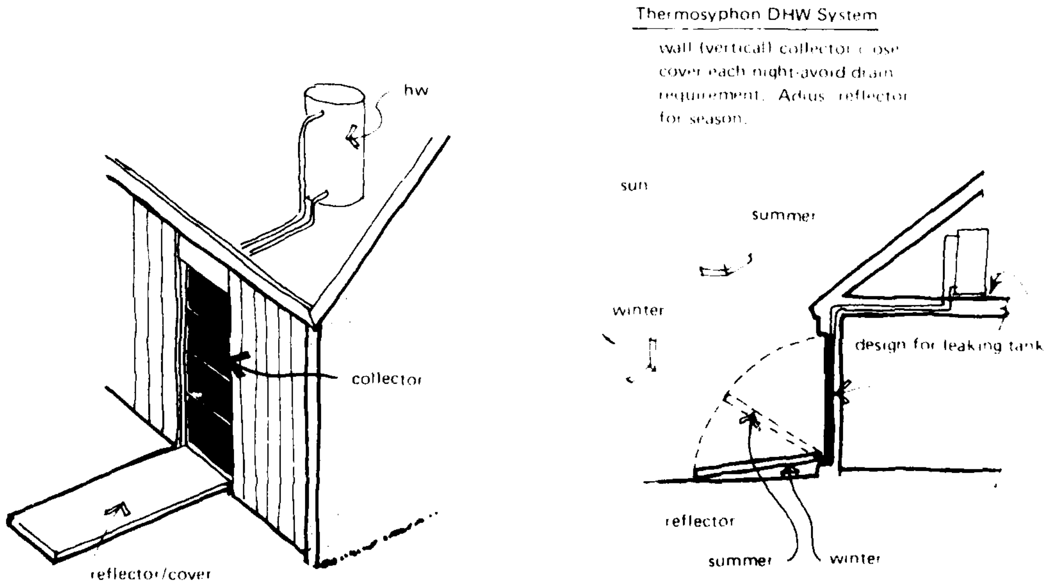
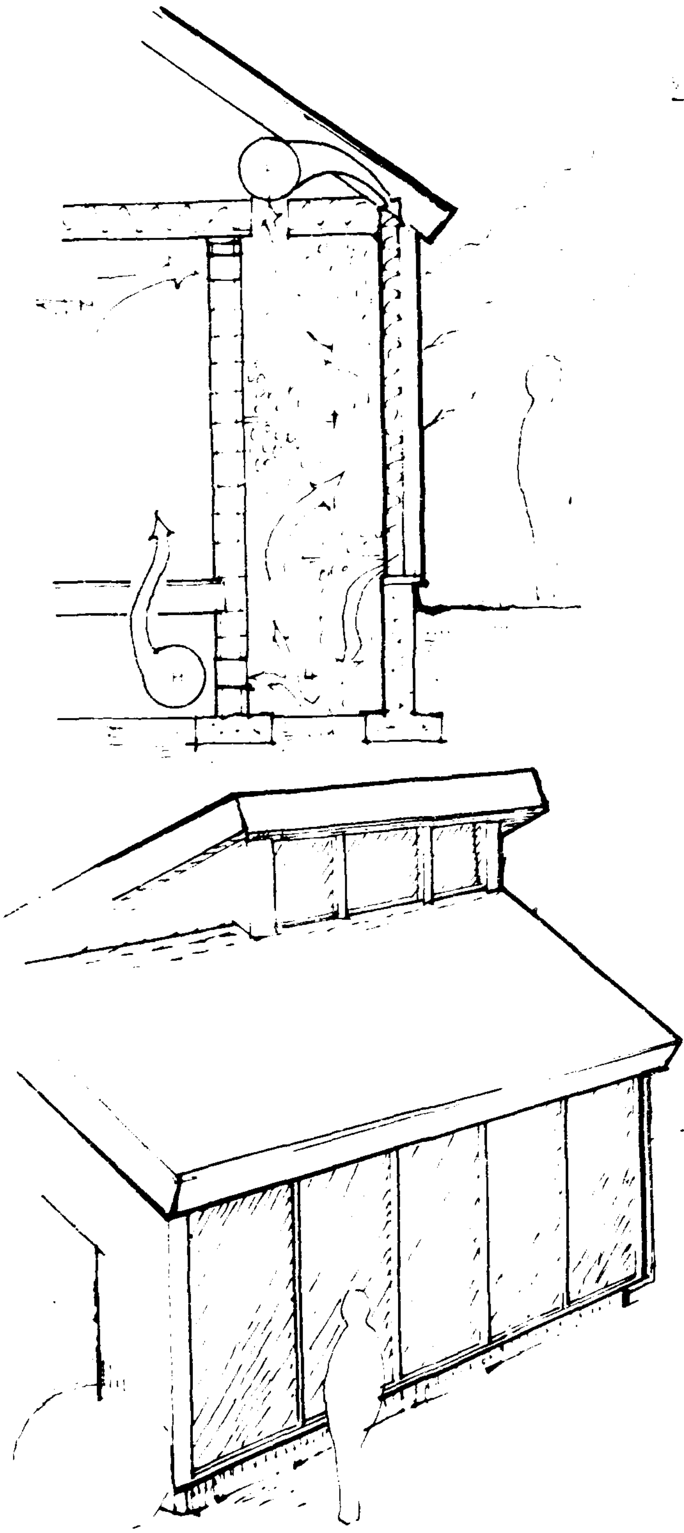


Figure 2-21. Vertical wall solar collector.



[illegible]

[illegible]

[illegible]

[illegible]

Takes [illegible] storage. [illegible] hot air from storage would be blown [illegible] house duct distrib. system. When storage temp. is [illegible] low, heater will come on to take over heating load.

Limited summer cooling can be obtained by [illegible] through rock bed.

Advantages

[illegible] heat loss from the air moving between storage and [illegible]

Reduces cost of [illegible] storage [illegible]

Problems

Some loss of efficiency by use of a vertical collector.

Potential shading problem from nearby trees or structures.

Figure 2-22. South wall solar collector with combined storage.

Retro Fit With Large Rock Storage

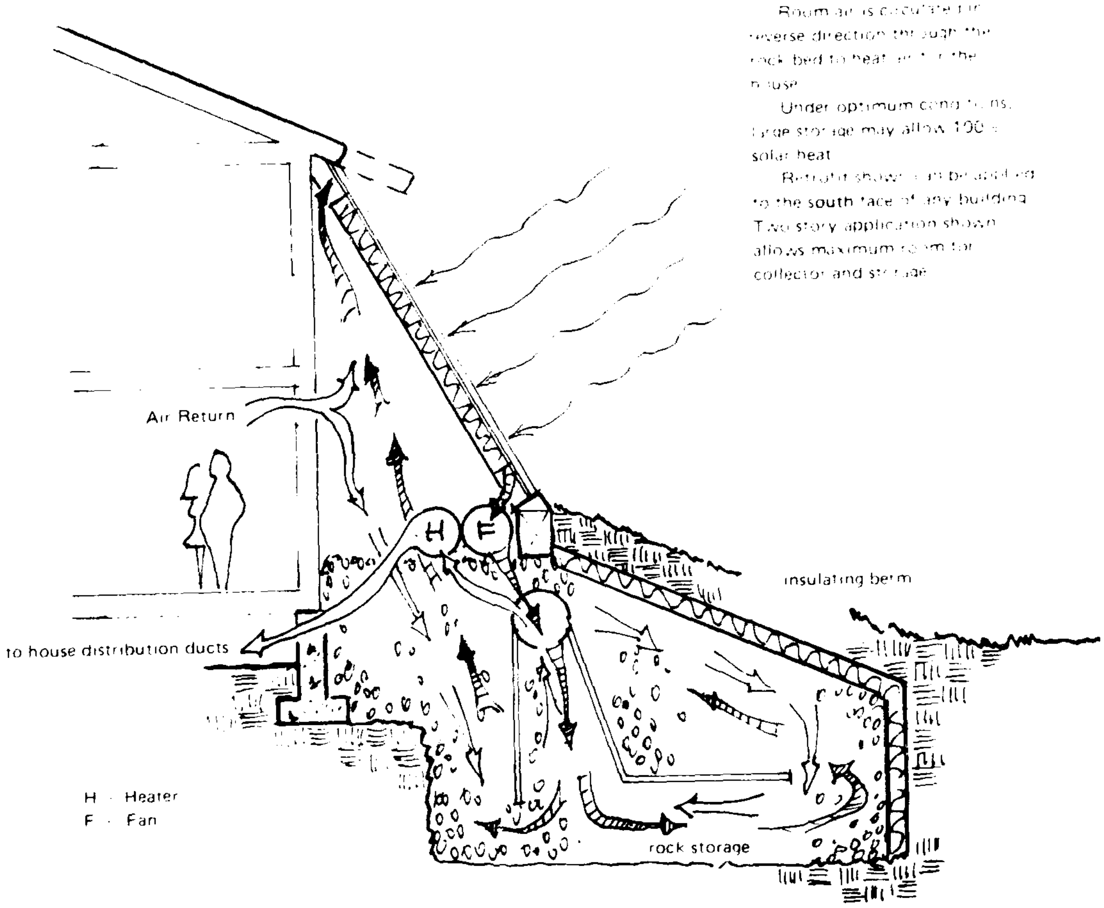
Operation

Hot air from [illegible] is circulated through [illegible] rock storage under insulating berm.

Room air is circulated in reverse direction through the rock bed to heat [illegible] the house.

Under optimum conditions, large storage may allow 100 [illegible] solar heat.

Retrofit shown can be applied to the south face of any building. Two story application shown allows maximum room for collector and storage.



Problems

Solar water heating would require a second heat exchanger at extra expense.

Ground position of collector may be shaded by trees or buildings.

High water table would interfere with storage.

Large amounts of rock may be expensive at some building sites.

Advantages

Excavation and backfill over the rock provides a low cost, well insulated containment for storage.

Figure 2-23. Retrofitted large rock bin storage.

(3) The best thickness of a Trombe wall is from 12 to 16 inches. The masonry should have a high density - at least 100 lb/ft^3. Thermocirculation vents can be used to increase daytime heating but will not increase nighttime minimums. Vents should have lightweight passive backdraft dampers or other means of preventing reverse flow at night. (Balcomb, undated)

(4) Two to three square feet of south-facing double glazing should be used for each Btu/°F-hr of additional thermal load (i.e., exclusive of the glazing). This will give 70% to 80% solar heating in northern New Mexico (Los Alamos) for a building kept within the range of 65°F to 75°F. See Balcomb (undated) for example of how to use this method.

(5) An easier to use rule in place of (4) is that given by Mazria (Mazria, 1979). For a well-insulated space in 40°N latitude in cold climates (outdoor temperature = 20°F to 30°F) the ratio of south glazing to floor area is in range 0.20 to 0.25 to maintain an average space temperature of 68°F over 24 hours (e.g., a 200 ft^2 floor space needs 40-50 ft^2 of south glazing). In temperate climates (35°F to 45°F outdoor temperature) use ratios in the range 0.11 to 0.17.*

(6) For Greenhouses: To determine solar gain: S = 1200 Btu/ft^2 of glazing per clear day, S = 700 Btu/ft^2 per average day. Double glaze only south wall. Insulate all opaque surfaces to R20, outside foundation to frost line to R10, minimize infiltration with caulking. Thermal mass = 5 gallons of water or 1-2/3 ft^3 of gravel per square foot of glazing. If storage is thermally isolated from greenhouse, air should be moved at 10 ft^3/min per square foot of glazing through the storage (McCullagh, 1978).

*Rules (4) or (5) have to be followed by rule (1) to estimate the storage required.

2.7 Solar Cooling Systems

The state-of-the-art of solar cooling has concentrated primarily on the developmental stages of systems in the last few years. Various methods have been researched, and some demonstrated, but only a few systems have been installed for other than research purposes. Solar cooling systems are attractive because cooling is most needed when solar energy is most available. If solar cooling can be combined with solar heating, the solar system can be more fully utilized and the economic benefits should increase. Solar cooling systems by themselves, however, are usually not economical at present fuel costs. Combining solar heating and cooling systems is not easy because of the different system requirements. This can best be understood by summarizing the different solar cooling techniques.

As with solar heating, the techniques for solar cooling consist of passive systems and active systems. The passive systems use some of the techniques discussed in Section 2.6 and further sources of information are Mazria (1979), Anderson (1976), and Bainbridge (1978). For active solar cooling systems the three most promising approaches are the heat actuated absorption machines, the Rankine cycle heat engine, and the desiccant dehumidification systems. A brief summary of these systems is given here and a more detailed explanation can be found in Merriam (1977) or other sources in the literature.

2.7.1 Absorption Cooling. Absorption cooling is the more commonly used method of solar cooling. An absorption refrigeration machine is basically a vapor-compression machine that accomplishes cooling by expansion of a liquid refrigerant under reduced pressure and temperature, similar in principle to an ordinary electrically operated vapor-compression air conditioner. Two refrigerant combinations have been used: lithium bromide and water and ammonia and water. There have been a number of proposed solid material absorption systems also. Figure 2-24 shows a typical lithium bromide absorption cooler. In the absorption cooler, heat is supplied to the generator in which a refrigerant is driven from a strong solution. The refrigerant is cooled in the condensor and allowed to expand through the throttling valve. The

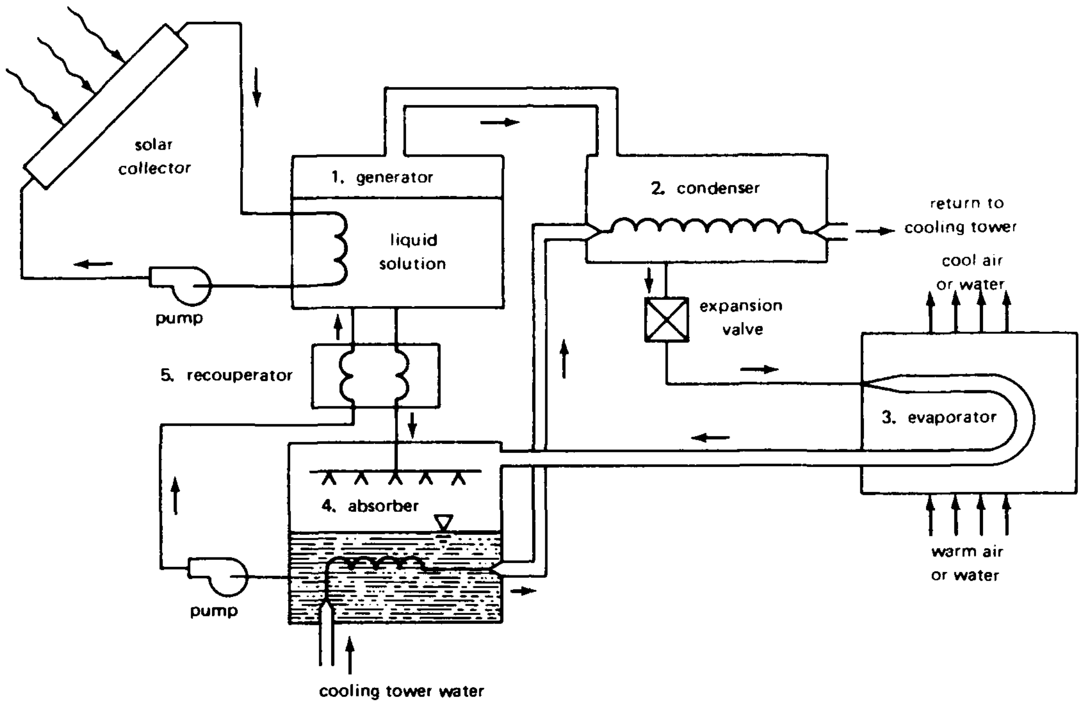


Figure 2-24. Schematic of lithium bromide absorption cooler.

cooled, expanded refrigerant receives heat in the evaporator to provide the desired cooling, after which the refrigerant is reabsorbed into the cool, weak solution in the absorber. The pressure of the resulting strong solution is increased by pumping and the solution is available to repeat the process.

The performance of the system is governed largely by the temperature difference between the generator and the condensor and absorber units. Since the generator temperatures in solar driven systems are only moderate, it is important to keep the condensor and absorber temperatures as low as possible.

The LiBr system is preferred over ammonia systems for solar energy applications because of the lower generator temperatures required. Permissible generator temperatures for a water cooled LiBr system range from 170°F to 210°F (76°C-99°C) compared to the 205°F to 248°F (95°C-120°C) temperatures required for a water cooled ammonia

absorption system. Most, if not all, of the commercially available absorption units use lithium bromide and water as the absorbent-refrigerant fluid pair. Because the lithium bromide will crystallize at the higher absorber temperatures associated with air cooling, these units must be water cooled. A prototype ammonia-water unit, amenable to direct air cooling, has been built by Lawrence Berkeley Laboratories.

A number of equipment requirements and limitations must be considered in the analysis and design of solar powered absorption systems. The first consideration involves the type of collector used. The temperatures required by absorption coolers are obtainable with flat plate collectors but at low collection efficiencies. Collection efficiency is improved with an increased number of glazings and with a selective surface, therefore, it may be cost effective to improve the collector rather than to simply oversize. If concentrating collectors are used (see Section 2.1.10.2), the associated higher costs and potentially increased maintenance for the tracking mechanism must be considered. In general, concentrating collectors operate at higher efficiency at these higher temperatures. However, the higher temperatures are usually not required to operate the space heating system. Therefore, the relative importance of the two thermal loads must be considered when selecting a system.

The second consideration involves the means of delivery of the heated fluid to the absorption cooler. Since, in many climates, the cooling load is simultaneous with and often proportional to the solar insolation, it may be desirable to allow the heated collector fluid to bypass the storage unit. Other climates may require a hot storage unit but one of considerably smaller size than the one used for heating purposes. The important requirement is that high temperatures be available during periods of heavy cooling load.

A third consideration deals with the problem of reduced efficiency of the absorption cooler under start up and transient conditions. Typical absorption coolers do not reach operating efficiency until after an hour or more of operation time. A machine which is cycled on and off regularly will have a drastically reduced average coefficient of performance when compared to a machine in steady state performance.

This problem has been overcome in at least one installation by the use of a cold storage unit (Beckman, 1977). The cold storage unit permits continuous operation of the absorption cooler and thus allows some reduction in the system and cooler size.

A fourth consideration is the need for some means of cooling the absorber and the condenser. A cooling tower or some other low temperature cooling system must be used to obtain reasonable performance. All of the commercially available units require a cooling tower which is another maintenance item. Current research is underway to develop units that do not have a separate cooling tower.

2.7.2 Rankine Cycle Heat Engine Cooling. Rankine cooling systems are still in development with only a few in operation (Anderson 1979; Barber 1975).

In these systems the shaft power produced by a heat engine drives the compressor in a conventional vapor compression-type cooling machine. The thermal energy input to the heat engine can be from a solar collector or from a solar collector and a fossil fuel combustor. The fossil fuel can supplement solar energy, or it can be used alone as the auxiliary energy supply when no solar energy is available. Alternatively, electricity can be used as the auxiliary energy supply by coupling an electric motor directly to the compressor shaft. Another option is a motor-generator using a heat engine for generating electricity when solar energy is available and there is little or no cooling load.

From state-of-the-art considerations, two types of fluid heat engines are primarily feasible in solar cooling units. In one type of engine, the working fluid cyclically changes phase from liquid to gas and back to liquid. The most widely used engine of this type operates on the Rankine cycle.

In the other type, the working fluid remains in the gaseous state. These engines operate on various cycles, including the Stirling and Brayton cycles. For relatively low thermal energy input temperatures (less than 400°F), Rankine cycle engines are superior in performance to gas cycle engines. At higher temperatures, gas cycle engines equal or better the performance of Rankine cycle engines.

Relatively low temperatures are attainable with state-of-the-art thermal solar collectors, so the heat engine-vapor compression development projects involve Rankine cycle engines.

In a Rankine cycle engine, fluid in the liquid state is pumped into a boiler where it is evaporated and possibly superheated by thermal energy. The vapor generated in the boiler is then expanded through a device such as a turbine, a piston-cylinder (reciprocating) expander, or a rotary vane expander. The expansion process lowers the temperature and pressure of the vapor, and effects a conversion of thermal energy into shaft work. The fluid leaves the expander either in the vapor phase or as a liquid-vapor mixture and flows into a condenser, where it returns to the liquid phase by giving the energy of condensation to cooling water or ambient air. This liquid is then pumped into the boiler, and the cycle is repeated.

In some systems under development, the same working fluid is used in both the Rankine engine and the vapor compression chiller, which permits the use of a common condenser and the elimination of special seals to maintain fluid separation in the expander-compressor unit (Scholten and Curran, 1979).

These systems have areas that need development in matching the solar heat engine with the mechanical compressor units of the cooling equipment. Since most compressors are designed for certain speed and torque inputs, the varying operation of a solar heat engine will probably reduce the overall COP of the unit. Also the solar heat engine is at high efficiency at high storage tank temperatures whereas the solar collectors are at low efficiency which will also affect the COP of the system. These systems are designed for large cooling load applications.

2.7.3 Desiccant Cooling (Scholten and Curran, 1979). The Rankine engine-vapor compression and the absorption cooling units operate on the basis of closed cycles - fixed amounts of working fluid are circulated within sealed equipment; the working fluids do not come in contact with the building air. Desiccant cooling systems, on the other hand, may be designed for open-cycle operation, since the only circulating fluids involved are air and water. The basic concept is to dehumidify air with a desiccant, evaporatively cool the dehumidified air, and regenerate the desiccant with solar-derived thermal energy.

Two basic open-cycle arrangements are feasible: the ventilation mode and the recirculation mode. In the ventilation mode, fresh air is continually introduced into the conditioned space. In the recirculation mode, exhaust air from the conditioned space is reconditioned and returned to the space. Figure 2-25 illustrates a ventilation system in which a solid desiccant material mounted on a slowly rotating wheel provides the basis for obtaining a cooling effect. The hot desiccant material absorbs moisture from incoming ventilation air and increases the dry-bulb temperature. This dry air stream is cooled in two steps. First, it is sensibly cooled by heat exchange with the building exhaust air. Then it is evaporatively cooled and partially rehumidified by contact with a water spray. The exhaust air from the building is evaporatively cooled to improve the performance of the heat exchanger. After being heated by heat exchange with the incoming air, the exhaust air is further heated by energy from the solar system and/or from an auxiliary energy source. The hot exhaust air passes through the desiccant material and desorbs moisture from it, thereby regenerating it for continuation of the process.

Desiccant systems have faced problems of high parasitic power and large space requirements relative to capacity. Because of their bulkiness, the systems may have primary application in the low capacity range (i.e., residential systems) if and when ways can be found to reduce parasitic power requirements to acceptable levels.

The Institute of Gas Technology (IGT) has been investigating design modifications in a prototype 3-ton system. AiResearch is developing a 1-1/2-ton desiccant cooling system around a radial flow design. Illinois Institute of Technology is developing a dehumidifier of a crossflow design that will provide more compact and efficient operation than previous designs. Zeopower is developing a unique closed cycle desiccant system in which the desiccant is integral with the collector.

2.7.4 Other Cooling Methods. Other methods, using solar heating equipment but not direct solar energy, should also be considered. These methods chill the thermal storage unit of the system during the night and use the chilled medium to provide the daily cooling load.

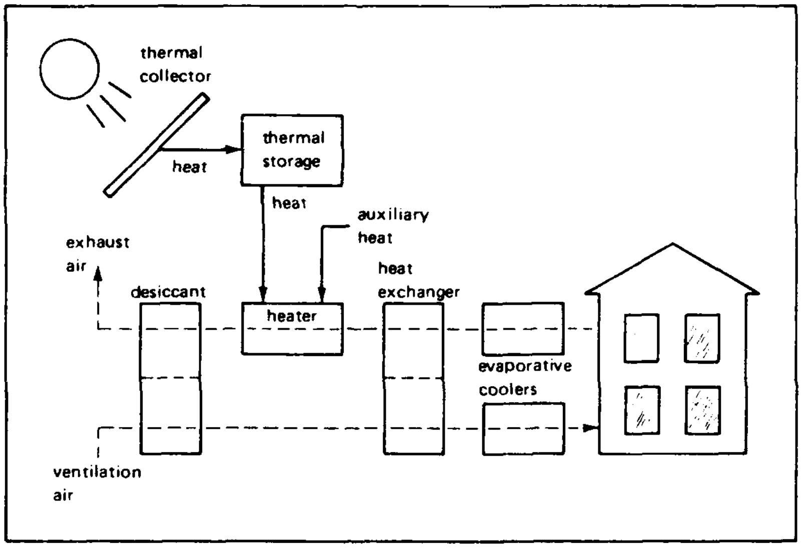


Figure 2-25. Schematic of solar desiccant cooling.

Methods of chilling the storage include radiation of the heat to the night sky and heat exchange with the night air cooled or uncooled by auxiliary means. The chilled storage is used directly, via heat exchange with the building air. Both rockbed and water storage are suitable since the only additional hardware required is that to route the fluid. A heat pump can be used during the day to cool the building and reject heat to the thermal storage unit. The thermal storage is then cooled by using the solar collectors for night sky radiation. From experimentation in Arizona, Bliss obtained a nightly heat rejection quantity of 360 Btu/night/ft^2 for a black cloth radiator. Analytical estimates can be obtained using an effective clear sky temperature of 25°F (14°C) lower than the ambient air temperature. The advantage of this system is that the same equipment (collectors and heat pump) can also be used for heating (Section 2.5.3). In systems with dual storage units, the heat pump transfers heat from one to the other - cooling the first and warming the second. The cool fluid in the first unit is circulated to the house while the concentrated heat in the second is discharged to the outdoors.

An evaporative cooler can be used coupled with a rockbed storage unit. Night air is evaporatively cooled and circulated through the rockbed to cool down the pebbles in the storage unit. During the day, warm air from the building can be cooled by passing it through the cool pebble bed.

The storage volume can also be cooled using a small refrigeration compressor. Most through-the-wall air conditioners use such compressors to cool the indoor air. This unit acts as the backup or auxiliary cooling system - analogous to the backup heating system. If operated only at night, its capacity can be as small as half that of an independently functioning unit and still meet peak cooling demands. Nighttime operation will be particularly wise if electric companies charge more for electricity during times of peak loads on hot summer afternoons. An even smaller compressor can be used if it operates continuously night and day - cooling the storage when not needed by the house (Anderson, 1976).

2.7.5 Estimating System Size. The sizing of cooling system components is dependent on hardware, climate, and economic constraints. The cooling unit must be sized so as to provide the maximum cooling load under conceivable adverse conditions of high humidity and low or erratic solar insolation.

The collection area required is dependent on the fraction of the cooling load to be provided by solar. Very large collector areas may be required for 100% solar cooling under adverse conditions of high humidity and low insolation. Although a detailed calculation method, as provided in the worksheets in the following sections for heating systems, is not available for solar cooling, an estimate of the required collector area can be made by the equation:

$$A = \frac{\text{Cooling load/COP}}{I_T \, \eta_{collect} \, \eta_{delivery}}$$

where			
	Cooling load	=	the portion of the total cooling load provided by solar - calculate using ASHRAE techniques or others
	COP	=	Coefficient of Performance of the cooling unit. COP is the ratio of heat energy removed to energy supplied from external sources. Manufacturing data is recommended for determining COP (3413 Btu = 1 kWh).
	I_T	=	average instantaneous solar insolation on collector surface (i.e., at tilt angle)
	$\eta_{collect}$	=	average collector efficiency under design conditions
	$\eta_{delivery}$	=	delivery efficiency which takes into account heat exchanger efficiency and thermal losses

In general, the collector area required to provide the majority of the cooling load is larger than the collector area of typically sized heating only systems. Collector areas for heat engine systems are larger than the areas for absorption cooling systems due to the thermal efficiency of the heat engine, which should be included in the preceding equation (Swindler, 1979).

2.8 System Controls

System controls are used to turn on a circulating pump or blower to the collector only when the sun is providing heat. Differential thermostats are commercially available (typically \$50 to \$150) to turn on the collector pump only when the collector plate temperature is a preset number (usually 20°F) hotter than the storage tank bottom temperature. A typical control strategy is shown in Figure 2-26 (Rho-Sigma, undated) and the hookup in Figure 2-27.

Differential thermostats are available with high temperature protection and low temperature (freeze) protection. Another type of control called proportional control is available. It is similar to the ON/OFF differential controller in operation. The difference is that the proportional controller changes the threshold ON and OFF points and controls

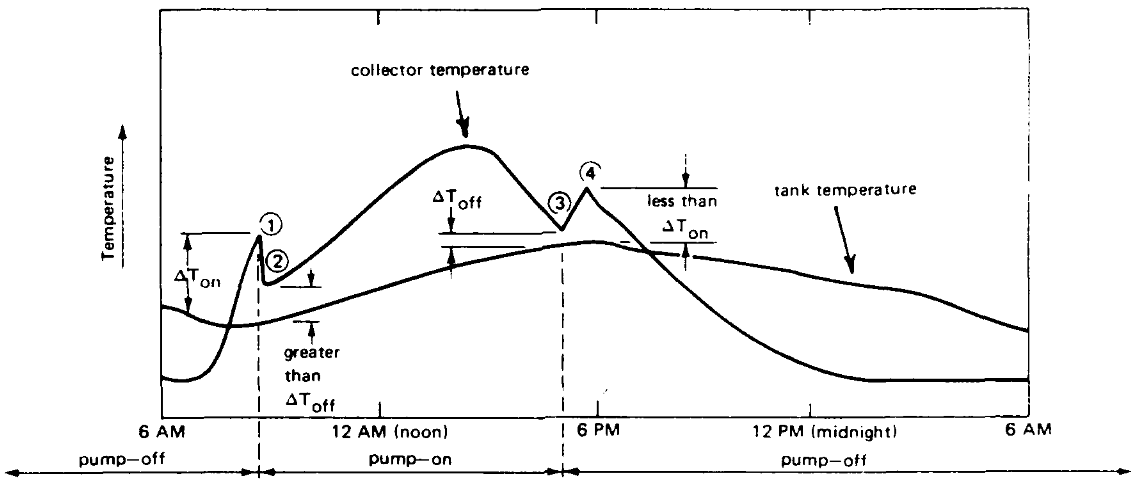


Figure 2-26. Control system strategy.

the flow such that less than full flow can be achieved if the sun is at less than full intensity. The advantage is that the proportional control can "turn on" the system when the other controller (the ON/OFF type) is waiting for more sun to become available. This is an advantage on cloudy days and early morning start ups. Overall system efficiency is increased slightly with the proportional control. These controls are more expensive and one such experiment at CEL has shown that proportional controls result in considerably more cycling of the pump motor which could shorten pump life. It is recommended that the control manufacturer be consulted on this point before a proportional control is used.

As the building requires heat, other controls must direct pumps or blowers to provide heat from the storage tank to the load. This control is the conventional thermostat. The same room thermostat may control the auxiliary heater; however, a delay timer or a two-step room thermostat must be incorporated into the auxiliary heater control circuit so that the auxiliary heat will not come on if heat is available from storage. Ten minutes has been suggested as a typical time delay before auxiliary heat comes on. Some manufacturers supply combination thermostat and solar system controls.

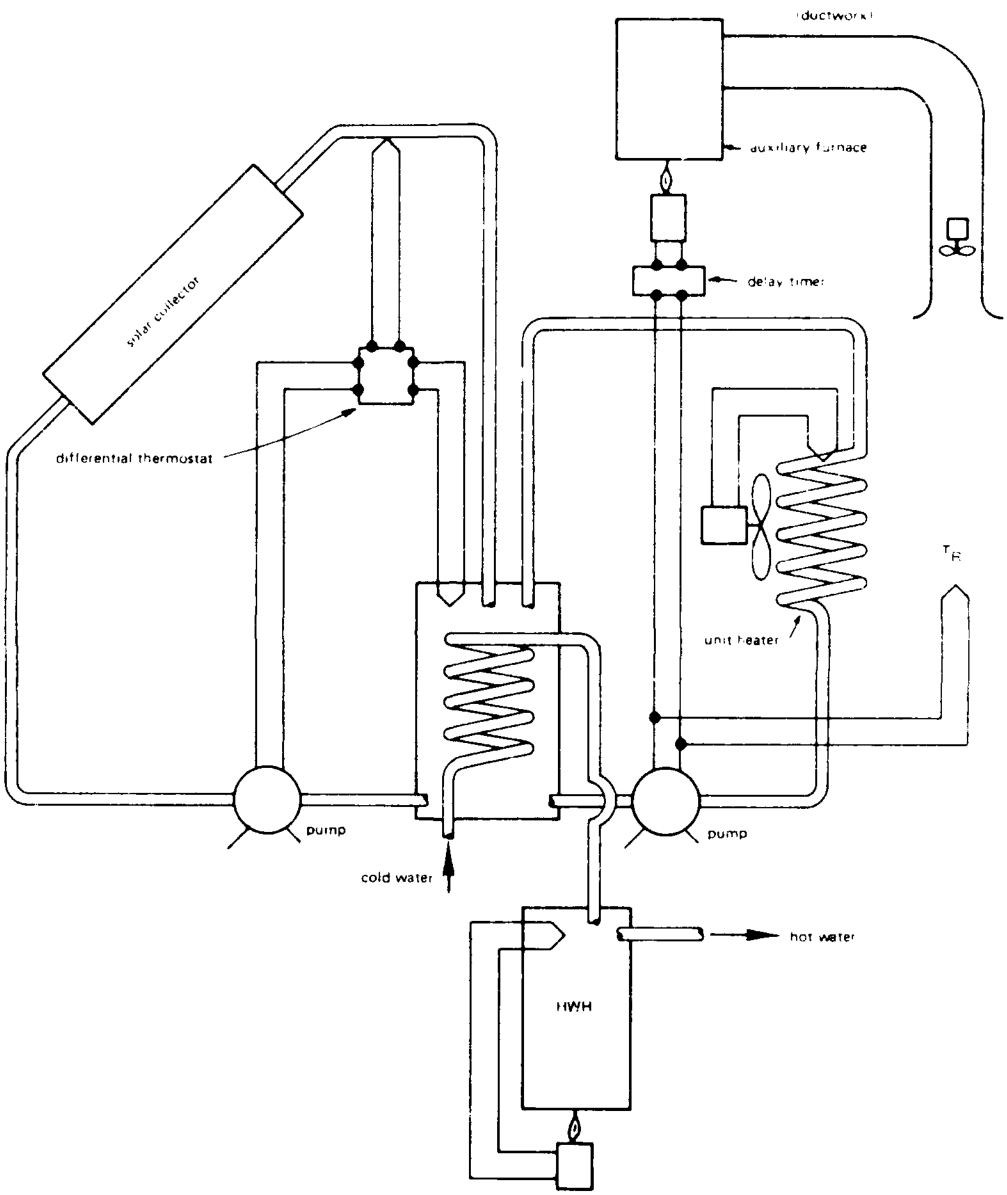


Figure 2-27. Control system for space and DHW heating.

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2.9 Piping, Pumps, Valves

2.9.1 Pipe and Header Sizing. Piping should be designed for low pressure drop. All exposed piping should be well insulated with approved weather-resistant insulation. Dielectric unions should be used at connections between dissimilar metals. Rubber or silicone hose used for connections must be of a high temperature type. Copper pipe is preferred to galvanized steel due to its longer life expectancy and relative ease of installation. Pipe sizing should be in accordance with recognized methods, but for most installations the following estimates are reasonable:

(1) For a single row of parallel collectors (see Figure 2-6) with "X" number of branches, 0.5 gpm flow per collector, water or 50% glycol as heat transfer fluid.

Up to 3 collectors	-	1/2-inch headers
4 to 7 collectors	-	3/4-inch headers
8 to 12 collectors	-	1-inch headers
13 to 18 collectors	-	1-1/4-inch headers
More than 19 collectors	-	1-1/2-inch or larger (size for each design)

(2) Same as above except collectors in a double row series-parallel arrangement (see Figure 2-6).

Up to 5 collector branches	-	1/2-inch headers
6 to 10 collector branches	-	3/4-inch headers
11 to 15 collector branches	-	1-inch headers
16 to 22 collector branches	-	1-1/4-inch headers
More than 23 collectors	-	1-1/2-inch or larger (size for each design)

2.9.2 Pumps and Collector Flowrate. Pumps are sized in accordance with recognized practices also. Since solar systems are nothing more than a combination of pipes, valves, and fittings it is possible to do a head loss calculation to determine the system head. Charts are available in standard fluid flow handbooks that give the friction losses or "equivalent length of feet in pipe" for various fittings and valves These are merely summed for the entire system.

The flowrate through the collector loop is determined by the maximum amount of energy which must be removed from the collector. This maximum is about 225 Btu/ft^2/hr. Often a manufacturer will specify the flowrate through his collector and this value should be used. If not, an estimate can be made by determining the flowrate necessary to remove the maximum amount of energy while minimizing the collector inlet temperature (to maintain high collector efficiency). The rule of thumb for this calculation is 0.015 to 0.020 gpm for each square foot of collector area for water. For other fluids this can be scaled by the value of the specific heat of the fluid as compared to water (C_p = 1 Btu/ lbm-°F).

Now that head loss and flowrate are determined, a pump can be selected by using the manufacturers' standard tables and graphs. For typical domestic hot water systems and space heating systems for a house for a family of four, the pumps are quite small, averaging 1/12 to 1/20 hp and costing about $125 each. In some closed loop systems, like a drain down system, pump sizes can be much larger due to the higher vertical "head" requirements.

If the water in the system is open to the atmosphere or if the water is to be used for drinking the pump should be made of bronze or stainless steel on all water-wetted surfaces to minimize corrosion. Pumps will have longer life if they are placed in low temperature parts of the water circuits. Pumps can be "staged" to give more flow or head. Two pumps in series will give the same flow against twice the head. Two pumps in parallel will give twice the flow at the same head. Two or more small circulator-type pumps are often cheaper than a single larger pump.

2.9.3 Valves. Valves, other than seasonal or emergency shut-off valves, should be electrically operated and located out of the weather or well protected. A vent must be provided at the high point in liquid systems to eliminate entrapped air and it should also serve as a vacuum breaker to allow draining of the system. To avoid multiple venting, systems should be piped to avoid having more than one high point. Check valves can be added to prevent thermally induced gravity circulation. A flow-check valve (used in the hydronic heating industry) will

also accomplish the same purpose. Mixing valves should be used to protect DHW systems from delivering water hotter than specified (usually 120°-140°F). Consideration should be given to energy conservation by lowering DWH temperature whenever possible. Often 105°-115°F will suffice if water is used only for showers and washing hands.

2.10 Other Considerations

2.10.1 Architectural. Solar collector arrangements should be studied to facilitate blending collector panels into the architecture of new or existing buildings. Shade trees must be so located as not to cast shadows on the collector. Other structures such as chimneys which can cast shadows should be carefully located to avoid shading of the collector. Experience of Florida installers indicates that if collectors are placed directly on the roof, the life of asphalt shingles under the collector may be reduced by up to 50%. This suggests that a small space should be left between the collector and the roof (although this has not yet been tested), or the collector should be built into the roof. In the latter case, the design must provide for simple glass replacement.

2.10.2 Reduction of Heat Losses. Reduction of heat losses is usually one of the most important steps in the design of a solar space heating system. It almost always costs less to super-insulate a building to reduce losses than to provide additional solar collector area to provide the extra heat. Installing 12 or more inches of insulation in the attic, insulating existing walls by injecting nonflammable foam (one manufacturer claims 30% reduction in total heat loss at cost $\$1.00/ft^2$ floor area), multiple glazing, and weatherstripping should all be evaluated for cost effectiveness versus a larger solar system. If the solar-augmented system is found to be cost competitive with a conventional system on a life cycle cost basis, then the cost effective amount of insulation will be the same for both the solar and conventional systems. Thus the solar system should not be charged for the cost of insulating the house.

2.10.3 <u>Maintenance and Accessibility</u>. Maintenance of glass will be minimized if vandalism can be reduced. Collectors of flat-roofed buildings may be shielded from the ground by a skirt around the roof perimeter. Locating the collector in the backyard area of residences rather than on a street-facing roof reduces probability of vandalism. Double strength glass for top surface is recommended in hail areas, and also provides protection from small stones. Still more protection is offered by a screen of 0.5-inch mesh stretched several inches above the collectors, but with some loss in collector efficiency (15%). Collectors and mounts must withstand expected wind and snow loads. Collector design should allow for rapid replacement of glass covers. Pumps, pipes, and controls should be reasonably accessible to allow repair or replacement. Water pumps should be located so that leakage does not cause serious damage.

3.0 DESIGN METHODS

There are three steps in the design of a solar system: determination of solar energy available per unit area of collector, determination of heating load, and sizing the collector for cost effectiveness. A series of worksheets (Section 3.13) has been prepared to facilitate the design process for liquid systems; see Section 3.12 for air systems. The worksheets should be duplicated as needed. The design method presented here is based substantially on the systems analysis done at the University of Wisconsin, Madison (Beckman, Klein, and Duffie, 1977; Klein, Beckman, and Duffie, 1976). The complex interaction between the components of a solar heating system has been reduced by means of computer simulation to a single parametric chart of F_IA_c versus F_LA_c with f as parameter (Figure 3-1), where F_I is a function of energy absorbed by the solar collector divided by building heating load, A_c is collector area, F_L is a function of solar collector heat losses divided by building heating load, and f is the fraction of building heating load supplied by solar heating. The requirement for advance knowledge of system temperatures has been eliminated by use of these heat balance ratios.

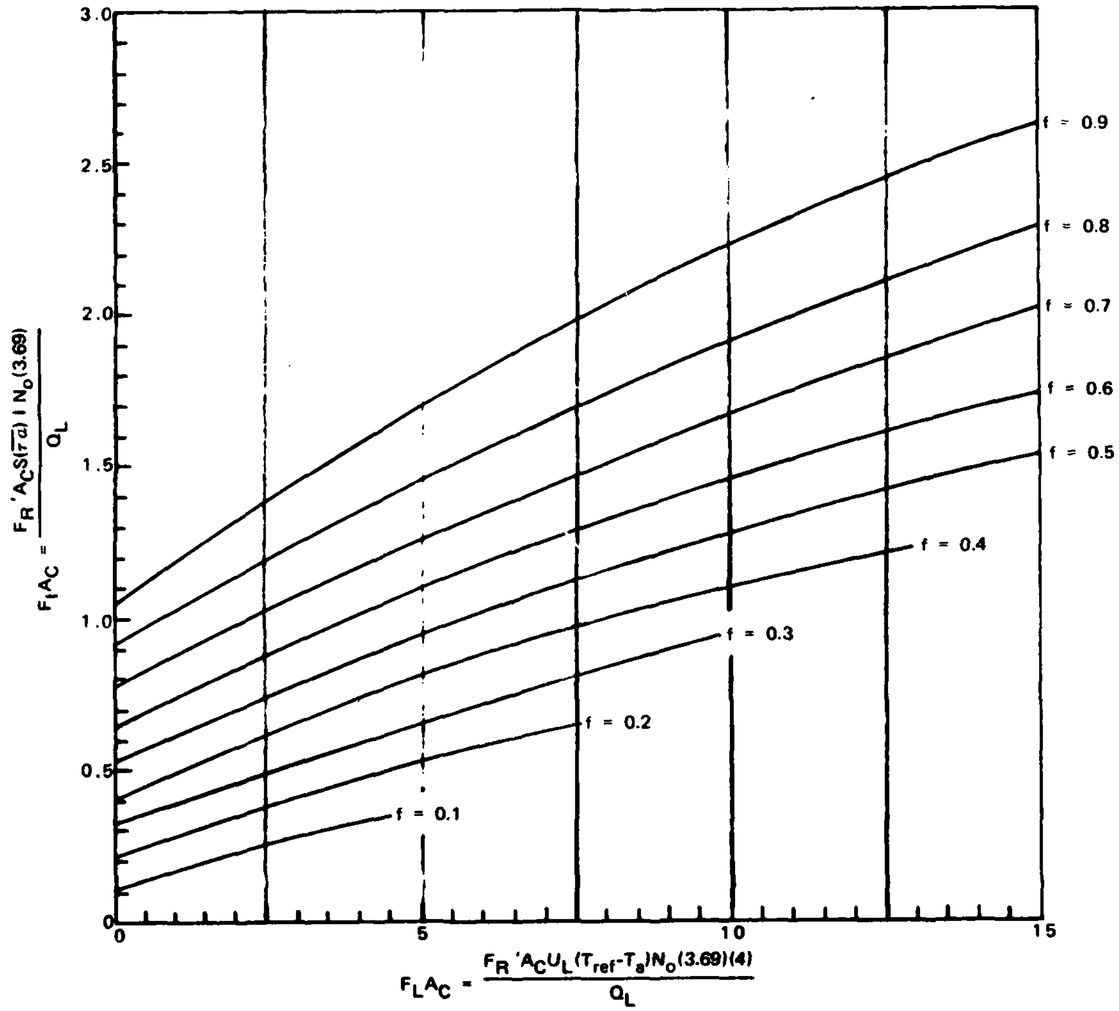

Figure 3-1. Fraction of space heating/DHW load supplied by solar energy (Beckman, Klein, and Duffie, 1977).

The method has been checked with computer simulations for the climates of Madison, Wisconsin; Blue Hill, Massachusetts; Charleston, South Carolina; Albuquerque, New Mexico; and Boulder, Colorado. The standard errors of the differences between the computer simulated and the values estimated by this method of $\bar{f}$ for the five locations were no greater than 0.014 (1.4% error); $\bar{f}$ is the yearly average of the monthly f. Eight years of data were used for the Madison, Wisconsin, case. This method, then appears to be sufficiently accurate for most applications and is a method widely used in the industry. It is the basis for an interactive computer program FCHART (Durlak, 1979b), hand calculator programs (Durlak, 1979b), and HUD reports (U.S. Dept HUD, 1977).

3.1 Job Summary - Worksheet A

Worksheet A is a summary sheet that shows the effect of collector size on savings-investment ratio (SIR). This is the final desired answer to the question of the design process: What size collector (and total system) gives greatest payback? If all SIRs are less than 1.0, then a solar system is not economical for the application at the conditions used in the design. A period of 25 years' fuel saving is used in calculations per NAVFAC P-442 as lifetime for utilities. Two methods are shown in Section 3.7. Solar systems can be designed to last this long. Computations completed on subsequent worksheets will be transferred to Worksheet A. Note that only the portion of conventional heating systems cost in excess of that normally required should be included in solar systems cost analysis. However, for budgetary purposes in new construction, then, the total solar system cost is the sum of the excess cost plus the previously excluded conventional system cost.

3.2 Solar Collector Parameters - Worksheet B

The purpose of Worksheet B is to gather the variables needed to calculate F_I and F_L (see paragraph 3.0). The first two parameters, $F_R(\tau\alpha)_n$ and F_RU_L represent the y intercept and slope, respectively, of

the η versus $\Delta T/I$ curve, Figure 2-7, applicable to the chosen collector. F_R is collector heat removal factor, τ is transmissivity of cover glazes, α is absorptivity of collector plate, U_L is overall collector heat loss coefficient. The y intercept ($F_R(\tau\alpha)_n$) and the slope ($F_R U_L$) can be read directly from Table 2-6 for a wide variety of collectors. The negative sign of the slope is ignored and the absolute value used. The y intercept is called efficiency intercept in Table 2-6. If a particular collector is not listed in Table 2-6, choose one with similar physical construction and use its value of slope and intercept. If necessary, slope and intercept can be calculated for a collector as shown in Duffie and Beckman (1974), but this is not recommended. If a manufacturer's brochure is used for obtaining test data, the slope will be a constant if the η versus $\Delta T/I$ curve is a straight line; however, if it is not a straight line, the slope to be used is the tangent to the curve in the expected range of $\Delta T/I$. The units of $F_R U_L$ must be Langleys/°F-day for use on subsequent worksheets.

The term $(\dot{m}C_p)_c/A_c$ is the unit heat capacity flowrate of working fluid through the collector, where m is flow of working fluid through the collector in lbm/hr, and C_p is specific heat of fluid in Btu/lbm F. The larger the flowrate, the lower will be the ΔT through collector and thus the higher will be the collector efficiency. A practical limit is reached at $(\dot{m}C_p)_c/A_c$ = 10 Btu/hr-ft²°F, so the design procedure is based on values of this order of magnitude. The latter figure may be taken as constant in calculating subsequent parameters.

The value for ε_c, effectiveness of the collector - tank heat exchanger, is based on manufacturer's data for the conditions of flow through the heat exchanger. If no heat exchanger is employed between the collector and the tank, then this term equals 1.0. For most heat exchangers, the effectiveness will vary from about 0.5 to 0.8. This is a relatively complex term to calculate and a procedure is given in U.S. Dept HUD (1977), pages A-34 to A-39. The simpler procedure is to ask a manufacturer for design information. Most manufacturers will provide this factor or calculate it. Be prepared to supply the following information:

- The physical characteristics of the two fluids in the heat exchanger
- The amount of heat to be transferred (Btu/hr)
- The flow rates (gal/min) on both sides of the heat exchanger
- The approach temperature difference defined as the difference between the temperatures of the hot fluid entering the heat exchanger and the heated fluid leaving the heat exchanger.

Alternatively, one could assume an effectiveness that is reasonable (say 0.7), then complete the following worksheets. If the design appears feasible, one could then go back and "refine" his estimate by consulting a manufacturer to be sure that a heat exchanger is available that will provide the assumed effectiveness. Such a heat exchanger might be employed if the working fluid were expensive, to reduce the amount of fluid required, or if it were desired to separate working fluid from potable water in a hot water storage tank (see Section 2.1.7 and Figure 2-5).

The term $(\dot{m}C_p)_c/(\dot{m}C_p)_{min}$ is the ratio of the heat capacity flow rates in the collector-tank heat exchanger. The subscript "c" refers to the collector flow stream; the subscript "min" refers to whichever of the two flowrates has the lesser value.

The ratio F_R'/F_R, line 6, Worksheet B, where F_R' is collector-tank heat exchanger efficiency factor is calculated from the equation

$$\frac{F_R'}{F_R} = \left\{1 + \left[F_R\, U_L \left(\frac{A_c}{(\dot{m}\, C_p)_c}\right)\right] \left[\frac{(\dot{m}\, C_p)_c}{\varepsilon_c (\dot{m}\, C_p)_{min}} - 1\right]\right\}^{-1}$$

using the factors previously developed. If there is to be no heat exchanger, then this ratio equals 1.0 for air and liquid systems.

Typically, if a heat exchanger is used with an effectiveness of 0.7 as assumed above and if the collector flow, $(\dot{m}C_p)_c$, is about the same as the other (storage) flow, $(\dot{m}C_p)_{min}$, as is often the case, then the

ratio F'_R/F_R is about 0.95 to 0.97 for a liquid system. This is not true for air systems which must be calculated. See U.S. Dept HUD (1977), pages A-26 and A-27 for further calculations of this parameter.

The next factor is $(\overline{\tau\alpha})/(\tau\alpha)_n$, line 7 Worksheet B, where the bar refers to an average value of $(\tau\alpha)$ and subscript "n" refers to the value of $(\tau\alpha)$ taken with the sun normal to the collector. This factor represents the variations in transmittance and absorptance due to changes in the sun angle during the day. Solar collectors are tested near solar noon, and there is a reduction in transmission of insolation at high angles of incidence which occur in early morning or late afternoon. This reduction (called incident angle modifier) may be available from the manufacturer of the collector since it is a parameter that is measured as part of a normal ASHRAE 93-77 performance test. If it is not available it may be taken as a constant as follows:

$$\frac{(\overline{\tau\alpha})}{(\tau\alpha)_n} = \begin{cases} 0.91 \text{ for two cover plates} \\ 0.93 \text{ for one cover plate} \end{cases}$$

In the final two parameters on Worksheet B, corrections for heat exchanger effectiveness (ε_c) and off-angle solar collection are made to the basic measured parameters to result in $F'_R(\tau\alpha)$ and $F'_R U_L$, which will be used in Worksheet D-1.

3.3 Load Calculations - Worksheet C-1

Worksheet C-1 is an aid to calculating the space heating and domestic hot water load for family housing. For other buildings use conventional methods of calculating load; computer programs are available for this. For existing buildings, heating load may be inferred from fuel bills, if available, see Example 1 (Section 4.1); or the Btu/ft^2 degree-day (dd) method of Worksheet C-1 may be used. Table 3-1 gives estimated Btu/ft^2 dd heat loss rates for various structural types used in family housing. If net heat loss rate is based on amount of fuel used, load is gross load and must be multiplied by furnace efficiency to get net heating load.

Table 3-1. Building Heat Loss Rates

Construction	Net Loss Coefficient (Btu/ft^2 degree day)
A. Brick veneer, 4-bedroom house, asphalt roof, storm windows, no insulation, 15 mph wind	15.3
B. Same as A, but with 3-1/2 inch batts in walls and attic	9.3
C. Same as B, but with 6 inches insulation in attic	8.7
D. Same as B, but with 12 inches insulation in attic	8.4
E. Stucco over frame, 4 bedroom house, shake roof, 3-inch insulation in attic only	14.1
F. Frame, 3 bedroom, heated basement, 3-1/2-inch batts in walls, 5-1/2-inch batts in ceiling	11.7

If Table 3-1 is not used, or if space heating loads for other types of buildings are desired the following sources give details on how to calculate loads:

(1) The ASHRAE Handbook of Fundamentals describes the basic method for calculating heat losses in a chapter entitled "Heating Load."

(2) U.S. Dept HUD (1977), page 16, section 3.3.1, describes a modified degree-day method for space heating loads.

Table 3-2 contains degree-day data for use in Worksheet C-1. This data is excerpted from U.S. NOAA (1968). Additional locations are available in U.S. NOAA (1968) and U.S. NAVFACENGCOM (1978). Hot water usage is calculated on Worksheet C-2 (see Section 3.4) and transferred to Worksheet C-1. Net DHW use is desired; if gross figure based on fuel usage is the starting point, then it must be multiplied by heater efficiency to get net DHW load. Utilization efficiency rather than an equipment efficiency should be used (see DM-3). Total net heating load is sum of space and DHW loads.

Table 3-2. Normal Total Heating Degree Days (Base 65°F)

State and Station	Jul	Aug	Sep	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May	Jun	Annual
ALABAMA													
Birmingham	0	0	6	93	363	555	592	462	363	108	9	0	2,551
Huntsville	0	0	12	127	426	663	694	557	434	138	19	0	3,070
Mobile	0	0	0	22	213	357	415	300	211	42	0	0	1,560
ALASKA													
Annette	242	208	327	567	738	899	949	837	843	648	490	321	7,069
Barrow	803	840	1035	1500	1971	2362	2517	2332	2468	1944	1445	957	20,174
Fairbanks	171	332	642	1203	1833	2254	2359	1901	1739	1068	555	222	14,279
ARIZONA													
Tucson	0	0	0	25	231	406	471	344	242	75	6	0	1,800
Yuma	0	0	0	0	148	319	363	228	130	29	0	0	1,217
ARKANSAS													
Little Rock	0	0	9	127	465	716	756	577	434	126	9	0	3,219
CALIFORNIA													
Fresno	0	0	0	78	339	558	586	406	319	150	56	0	2,492
Long Beach	0	0	12	40	156	288	375	297	267	168	90	18	1,711
Los Angeles	28	22	42	78	180	291	372	302	288	219	158	81	2,061
Oakland	53	50	45	127	309	481	527	400	353	255	180	90	2,870
Sacramento	0	0	12	81	363	577	614	442	360	216	102	6	2,773
San Diego	6	0	15	37	123	251	313	249	202	123	84	36	1,439
San Francisco	81	78	60	143	306	462	508	392	363	279	214	126	3,015
Santa Maria	99	93	96	146	270	391	459	370	363	282	233	165	2,967
COLORADO													
Denver	6	9	117	428	819	1035	1132	938	887	558	288	66	6,283
Grand Junction	0	0	30	313	786	1113	1209	907	729	387	146	21	5,641
CONNECTICUT													
Bridgeport	0	0	66	307	615	986	1079	966	853	510	208	27	5,617
New Haven	0	12	87	347	648	1011	1097	991	871	543	245	45	5,897

(continued)

Table 3-2. Continued

State and Station	Jul	Aug	Sep	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May	Jun	Annual
FLORIDA													
Apalachicola	0	0	0	16	153	319	347	260	180	33	0	0	1,308
Jacksonville	0	0	0	12	144	310	332	246	174	21	0	0	1,239
Key West	0	0	0	0	0	28	40	31	9	0	0	0	108
Miami Beach	0	0	0	0	0	40	56	36	9	0	0	0	141
Pensacola	0	0	0	19	195	353	400	277	183	36	0	0	1,463
GEORGIA													
Atlanta	0	0	18	127	414	626	639	529	437	168	25	0	2,983
Columbus	0	0	0	87	333	543	552	434	338	96	0	0	2,383
IDAHO													
Boise	0	0	132	415	792	1017	1113	854	722	438	245	81	5,809
Idaho Falls 46W	16	34	270	623	1056	1370	1538	1249	1085	651	391	192	8,475
ILLINOIS													
Chicago	0	0	81	326	753	1113	1209	1044	890	480	211	48	6,155
Springfield	0	0	72	291	696	1023	1135	935	769	354	136	18	5,429
INDIANA													
Indianapolis	0	0	90	316	723	1051	1113	949	809	432	177	39	5,699
IOWA													
Des Moines	0	9	99	363	837	1231	1398	1165	967	489	211	39	6,808
KANSAS													
Dodge City	0	0	33	251	666	939	1051	840	719	354	124	9	4,986
Topeka	0	0	57	270	672	980	1122	893	722	330	124	12	5,182
KENTUCKY													
Lexington	0	0	54	239	609	902	946	818	685	325	105	0	4,683
Louisville	0	0	54	248	809	890	930	818	682	315	105	9	4,660

(continued)

Table 3-2. Continued

State and Station	Jul	Aug	Sep	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May	Jun	Annual
LOUISIANA													
Lake Charles	0	0	0	19	210	341	381	274	195	39	0	0	1,459
New Orleans	0	0	0	19	192	322	363	258	192	39	0	0	1,385
MAINE													
Portland	12	53	195	508	807	1215	1339	1182	1042	675	372	111	7,511
MARYLAND													
Baltimore	0	0	48	264	585	905	936	820	679	327	90	0	4,654
MASSACHUSETTS													
Boston	0	9	60	316	603	983	1088	972	846	513	208	36	5,634
MICHIGAN													
Detroit (City)	0	0	87	360	738	1088	1181	1058	936	522	220	42	6,232
Sault Ste. Marie	96	105	279	580	951	1367	1525	1380	1277	810	477	201	9,048
MINNESOTA													
Saint Cloud	28	47	225	549	1065	1500	1702	1445	1221	666	326	105	8,879
MISSISSIPPI													
Jackson	0	0	0	65	315	502	546	414	310	87	0	0	2,239
MISSOURI													
Columbia	0	0	54	251	651	967	1076	874	716	324	121	12	5,046
St. Louis	0	0	60	251	627	936	1026	848	704	312	121	15	4,900
MONTANA													
Great Falls	28	53	258	543	921	1169	1349	1154	1063	642	384	186	7,750
NEBRASKA													
Lincoln	0	6	75	301	726	1066	1237	1016	834	402	171	30	5,864
Omaha	0	12	105	357	828	1175	1355	1126	939	465	208	42	6,612

(continued)

Table 3-2. Continued

State and Station	Jul	Aug	Sep	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May	Jun	Annual
NEVADA													
Ely	28	43	234	592	939	1184	1308	1075	977	672	456	225	7,733
Las Vegas	0	0	0	78	387	617	688	487	335	111	6	0	2,709
Reno	43	87	204	490	801	1026	1073	823	729	510	357	189	6,332
NEW JERSEY													
Trenton	0	0	57	264	576	924	989	885	753	399	121	12	4,980
NEW MEXICO													
Albuquerque	0	0	12	229	642	868	930	703	595	288	81	0	4,348
NEW YORK													
J. F. Kennedy Intl	0	0	36	248	564	933	1029	935	815	480	167	12	5,219
NORTH CAROLINA													
Cape Hatteras	0	0	0	78	273	521	580	518	440	177	25	0	2,612
Greensboro	0	0	33	192	513	778	784	672	552	234	47	0	3,805
NORTH DAKOTA													
Bismarck	34	28	222	577	1083	1463	1708	1442	1203	645	329	117	8,851
OHIO													
Cleveland	9	25	105	384	738	1088	1159	1047	918	552	260	66	6,351
Columbus	0	6	84	347	714	1039	1088	949	809	426	171	27	5,660
OKLAHOMA													
Oklahoma City	0	0	15	164	498	766	868	664	527	189	34	0	3,725
OREGON													
Astoria	146	130	210	375	561	679	753	622	636	480	363	231	5,186
Portland	25	28	114	335	597	735	825	644	586	396	245	105	4,635
PENNSYLVANIA													
Philadelphia	0	0	0	291	621	964	1014	890	744	390	115	12	5,101
RHODE ISLAND													
Providence	0	16	96	372	660	1023	1110	988	868	534	236	51	5,954

(continued)

Table 3-2. Continued

State and Station	Jul	Aug	Sep	Oct	Nov	Dec	Jan	Feb	Mar	Apr	May	Jun	Annual
SOUTH CAROLINA													
Charleston	0	0	0	59	282	471	487	389	291	54	0	0	2,033
SOUTH DAKOTA													
Rapid City	22	12	165	481	897	1172	1333	1145	1051	615	326	126	7,345
TENNESSEE													
Nashville	0	0	30	158	495	732	778	644	512	189	40	0	3,578
TEXAS													
Brownsville	0	0	0	0	66	149	205	106	74	0	0	0	600
Corpus Christi	0	0	0	0	120	220	291	174	109	0	0	0	914
Dallas	0	0	0	62	321	524	601	440	319	90	6	0	2,363
El Paso	0	0	0	84	414	648	685	445	319	105	0	0	2,700
San Antonio	0	0	0	31	207	363	428	286	195	39	0	0	1,549
UTAH													
Salt Lake City	0	0	81	419	849	1082	1172	910	763	459	233	84	6,052
VERMONT													
Burlington	28	65	207	539	891	1349	1513	1333	1187	714	353	90	8,269
VIRGINIA													
Norfolk	0	0	0	136	408	698	738	655	533	216	37	0	3,421
WASHINGTON													
Seattle	50	47	129	329	543	657	738	599	577	396	242	117	4,424
WEST VIRGINIA													
Charleston	0	0	63	254	591	865	880	770	648	300	96	9	4,476
WISCONSIN													
Green Bay	28	50	174	484	924	1333	1494	1313	1141	654	335	99	8,029
Madison	25	40	174	474	930	1330	1473	1274	1113	618	310	102	7,863
Milwaukee	43	47	174	471	876	1252	1376	1193	1054	642	372	135	7,635
WYOMING													
Cheyenne	19	31	210	543	924	1101	1228	1056	1011	672	381	102	7,278
Lander	6	19	204	555	1020	1299	1417	1145	1017	654	381	153	7,870

3.4 Demand Calculations - DHW - Worksheet C-2

Worksheet C-2 summarizes DHW demand determined by conventional methods: manual DM-3, chapter 1. Table 2-9 (Section 2.3) gives hot water demands for various buildings. Table 3-3 gives temperature of water mains for various locations. Net DHW load, Q_d x N_o, (Q_d is Btu/day hot water demand) is transferred to column (W), Worksheet C-1, if a combined space heating DHW system is being designed. If hot water demand is calculated from fuel bills, a gross figure is obtained, which is entered in column (U), Worksheet C-1. Net demand = gross x η_w, where η_w = heater utilization efficiency.

3.5 Monthly Solar Collection Parameters - Worksheet D-1

Figures for Q_L, total heat load per month, are transferred to Worksheet D-1. Solar insolation, I, and slope factor, S, are obtained from Table 1-1 and Figure 3-2, respectively, for the location and latitude. If measured I for the location is available for several years, then the average of this data should be used. The slope factor corrects insolation data from the horizontal at which the insolation data were taken, to the tilt angle of the collector. If the tilt angle is latitude plus 10 degrees, then Figure 3-2 may be used for S. For deviation from "latitude plus 10 degrees," see Duffie and Beckman (1974). These calculations apply to south-facing collectors; no correction is needed for collectors facing up to 10 degrees east or west of south. The air temperature, T_a, is the average daily temperature taken from local records or the Climatic Atlas of the United States (U.S. NOAA, 1968) which is excerpted in Table 3-4. The factor (T_{ref} - T_a) accounts for the effects of ambient air temperature changes on collector heat losses. Then the parameters F_I and F_L may be calculated. Special care should be taken that units are consistent, so that F_I and F_L will have units of ft^{-2}. The factors (3.69) and (4.0) are necessary to achieve this. The factor (4.0) converts hours of sunlight per day (6) to hours per day (24).

Table 3-3. DHW Temperature at Source in Selected Locations (U.S. Dept HUD, 1977)

[Temperatures in °F]

City	Source[a]	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
1. Phoenix	Ri, Re, W	48	48	50	52	57	59	63	75	79	69	59	54
2. Miami	W	70	70	70	70	70	70	70	70	70	70	70	70
3. Los Angeles	Ri, W	50	50	54	63	68	73	74	76	75	69	61	55
4. Albuquerque	W	72	72	72	72	72	72	72	72	72	72	72	72
5. Las Vegas	W	73	73	73	73	73	73	73	73	73	73	73	73
6. Denver	Ri	39	40	43	49	55	60	63	64	73	56	45	37
7. Ft. Worth	L	56	49	57	70	75	81	79	83	81	72	56	46
8. Nashville	Ri	46	46	53	66	63	69	71	75	75	71	58	53
9. Washginton, DC	Ri	42	42	52	56	63	67	67	78	79	68	55	56
10. Salt Lake City	W, C	35	37	38	41	43	47	53	52	48	43	38	37
11. Seattle	Ri	39	37	43	45	48	57	60	68	66	57	48	43
12. Boston	Re	32	36	39	52	58	71	74	67	60	56	48	45
13. Chicago	L	32	32	34	42	51	57	65	67	62	57	45	35
14. New York City	Re	36	35	36	39	47	54	58	60	61	57	48	45

[a]Abbreviations: C - Creek, L - Lake, Re - Reservoir, Ri - River, W - Well.

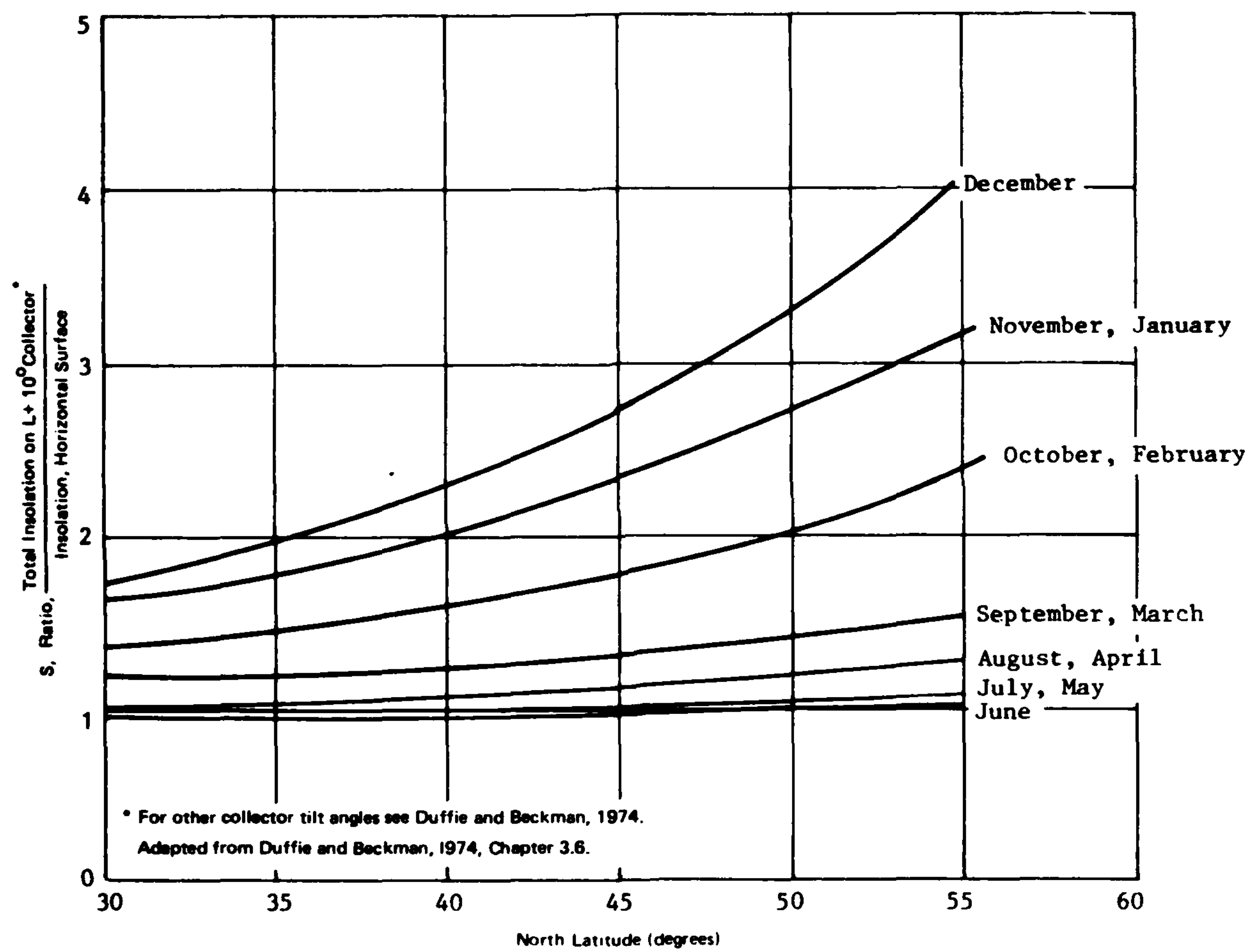

Figure 3-2. Slope factor, S, for use on Worksheet D-1 (average over one day).

Table 3-4. Average Daytime Ambient Temperatures (°F) (U.S. NOAA, 1968)

State and Station	Latitude	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
ALASKA													
Annette Island	55°01'	35.8	37.5	39.7	44.4	51.0	56.2	38.6	59.8	54.8	48.2	41.5	37.4
Barrow	71°20'	-13.2	-15.9	-12.7	2.1	20.5	35.4	41.6	40.0	31.7	18.6	2.6	-8.6
Fairbanks	64°49'	-7.0	0.3	13.0	32.2	50.5	62.4	63.8	58.3	47.1	29.6	5.5	-6.6
ARIZONA													
Phoenix	33°26'	54.2	58.8	64.7	72.2	80.8	89.2	94.6	92.5	87.4	75.8	63.6	56.7
Tucson	32°07'	53.7	57.3	62.3	69.7	78,0	87.0	90.1	87.4	84.0	73.9	62.5	56.1
CALIFORNIA													
Davis	38°33'	47.6	52.1	56.8	63.1	69.6	75.7	81.0	79.4	76.7	67.8	57.0	48.7
Fresno	36°46'	47.3	53.9	59.1	65.6	73.5	80.7	87.5	84.9	78.6	68.7	57.3	48.9
Inyokern	35°39'	47.3	53.9	59.1	65.6	73.5	80.7	87.5	84.9	78.6	68.7	57.3	48.9
Los Angeles	34°03'	57.9	59.2	61.8	64.3	67.6	70.7	75.8	76.1	74.2	69.6	65.4	60.2
Riverside	33°57'	55.3	57.0	60.6	65.0	69.4	74.0	81.0	81.0	78.5	71.0	63.1	57.2
Santa Maria	34°54'	54.1	55.3	57.6	59.5	61.2	63.5	65.3	65.7	65.9	64.1	60.8	56.1
COLORADO													
Grand Junction	39°07'	26.9	35.0	44.6	55.8	66.3	75.7	82.5	79.6	71.4	58.3	42.0	31.4
Grand Lake	40°15'	18.5	23.1	28.5	39.1	48.7	56.6	62.8	61.5	55.5	45.2	30.3	22.6
DISTRICT OF COLUMBIA	38°51'	38.4	39.6	48.1	57.5	67.7	76.2	79.9	77.9	72.2	60.9	50.2	40.2
FLORIDA													
Apalachicola	29°45'	57.3	59.0	62.9	69.5	76.4	81.8	83.1	83.1	80.6	73.2	63.7	58.5
Gainesville	29°39'	62.1	63.1	67.5	72.8	79.4	83.4	83.8	84.1	82.0	75.7	67.2	62.4
Miami	25°47'	71.6	72.0	73.8	77.0	79.9	82.9	84.1	84.5	83.3	80.2	75.6	72.6
Tampa	27°55'	64.2	65.7	68.8	74.3	79.4	83.0	84.0	84.4	82.9	77.2	69.6	65.5
GEORGIA													
Atlanta	33°39'	47.2	49.6	55.9	65.0	73.2	80.9	82.4	81.6	77.4	66.5	54.8	47.7
Griffin	33°15'	48.9	51.0	59.1	66.7	74.6	81.2	83.0	82.2	78.4	68.0	57.3	49.4

(continued)

Table 3-4. Continued

State and Station	Latitude	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
IDAHO Boise	43°34'	29.5	36.5	45.0	53.5	62.1	69.3	79.6	77.2	66.7	56.3	42.3	33.1
ILLINOIS Lemont	41°40'	28.9	30.3	39.5	49.7	59.2	70.8	75.6	74.3	67.2	57.6	43.0	30.6
INDIANA Indianapolis	39°44'	31.3	33.9	43.0	54.1	64.9	74.8	79.6	77.4	70.6	59.3	44.2	33.4
KANSAS Dodge City	37°46'	33.8	38.7	46.5	57.7	66.7	77.2	83.8	82.4	73.7	61.7	46.5	36.8
KENTUCKY Lexington	38°02'	36.5	38.8	47.4	57.8	67.5	76.2	79.8	78.2	72.8	61.2	47.6	38.5
LOUISIANA Lake Charles	30°13'	55.3	58.7	63.5	70.9	77.4	83.4	84.8	85.0	81.5	73.8	62.6	56.9
MAINE Portland	43°39'	23.7	24.5	34.4	44.8	55.4	65.1	71.1	69.7	61.9	51.8	40.3	28.0
MASSACHUSETTS Boston	42°22'	31.4	31.4	39.9	49.5	60.4	69.8	74.5	73.8	66.8	57.4	46.6	34.9
MICHIGAN Sault Ste. Marine	46°28'	16.3	16.2	25.6	39.5	52.1	61.6	67.3	66.0	57.9	46.8	33.4	21.9
MINNESOTA St. Cloud	45°35'	13.6	16.9	29.8	46.2	58.8	68.5	74.4	71.9	62.5	50.2	32.1	18.3
MISSOURI Columbia	38°58'	32.5	36.5	45.9	57.7	66.7	75.9	81.1	79.4	71.9	61.4	46.1	35.8
MONTANA Great Falls	47°29'	25.4	27.6	35.6	47.7	57.5	64.3	73.8	71.3	60.6	51.4	38.0	29.1

(continued)

Table 3-4. Continued

State and Station	Latitude	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
NEBRASKA													
Lincoln	40°51'	27.8	32.1	42.4	55.8	65.8	76.0	82.6	80.2	71.5	59.9	43.2	31.8
NEVADA													
Ely	39°17'	27.3	32.1	39.5	48.3	57.0	65.4	74.5	72.3	63.7	52.1	39.9	31.1
Las Vegas	36°05'	47.5	53.9	60.3	69.5	78.3	88.2	95.0	92.9	85.4	71.7	57.8	50.2
NEW JERSEY													
Seabrook	39°30'	39.5	37.6	43.9	54.7	64.9	74.1	79.8	77.7	69.7	61.2	48.5	39.3
NEW MEXICO													
Albuquerque	35°03'	37.3	43.3	50.1	59.6	69.4	79.1	82.8	80.6	73.6	62.1	47.8	39.4
NEW YORK													
New York City	40°46'	35.0	34.9	43.1	52.3	63.3	72.2	76.9	75.3	69.5	59.3	48.3	37.7
NORTH CAROLINA													
Greensboro	36°05'	42.0	44.2	51.7	60.8	69.9	78.0	80.2	78.9	73.9	62.7	51.5	43.2
NORTH DAKOTA													
Bismarck	46°47'	12.4	15.9	29.7	46.6	58.6	67.9	76.1	73.5	61.6	49.6	31.4	18.4
OHIO													
Cleveland	41°24'	30.8	30.9	39.4	50.2	62.4	72.7	77.0	75.1	68.5	57.4	44.0	32.8
Columbus	40°00'	32.1	33.7	42.7	53.5	64.4	74.2	78.0	75.9	70.1	58.0	44.5	34.0
OKLAHOMA													
Oklahoma City	35°24'	40.1	45.0	53.2	63.6	71.2	80.6	85.5	85.4	77.4	66.5	52.2	43.1
OREGON													
Astoria	46°12'	41.3	44.7	46.9	51.3	55.0	59.3	62.6	63.6	62.2	55.7	48.5	43.9
PENNSYLVANIA													
State College	40°48'	31.3	31.4	39.8	51.3	63.4	71.8	75.8	73.4	66.1	55.6	43.2	32.6

(continued)

Table 3-4. Continued

State and Station	Latitude	Jan	Feb	Mar	Apr	May	Jun	Jul	Aug	Sep	Oct	Nov	Dec
RHODE ISLAND													
Newport	41°29'	29.5	32.0	39.6	48.2	58.6	67.0	73.2	72.3	66.7	56.2	46.5	34.4
SOUTH CAROLINA													
Charleston	32°54'	53.6	55.2	60.6	67.8	74.8	80.9	82.9	82.3	79.1	69.8	59.8	54.0
SOUTH DAKOTA													
Rapid City	44°09'	24.7	27.4	34.7	48.2	58.3	67.3	76.3	75.0	64.7	52.9	38.7	29.2
TEXAS													
Brownsville	25°55'	63.3	66.7	70.7	76.2	81.4	85.1	86.5	86.9	84.1	78.9	70.7	65.2
El Paso	31°48'	47.1	53.1	58.7	67.3	75.7	84.2	84.9	83.4	78.5	69.0	56.0	48.5
Fort Worth	32°50'	48.1	52.3	59.8	68.8	75.9	84.0	87.7	88.6	81.3	71.5	58.8	50.8
San Antonio	29°32'	53.7	58.4	65.0	72.2	79.2	85.0	87.4	87.8	82.6	74.7	63.3	56.5
TENNESSEE													
Nashville	36°07'	42.6	45.1	52.9	63.0	71.4	80.1	83.2	81.9	76.6	65.4	52.3	44.3
UTAH													
Salt Lake City	40°46'	29.4	36.2	44.4	53.9	63.1	71.7	81.3	79.0	68.7	57.0	42.5	34.0
WASHINGTON													
Seattle	47°27'	38.9	42.9	46.9	51.9	58.1	62.8	67.2	66.7	61.6	54.0	45.7	41.5
WISCONSIN													
Madison	43°08'	21.8	24.6	35.3	49.0	61.0	70.9	76.8	74.4	65.6	53.7	37.8	25.4
WYOMING													
Lander	42°48'	20.2	26.3	34.7	45.5	56.0	65.4	74.6	72.5	61.4	48.3	33.4	23.8

3.6 Fraction of Load Supplied by Solar Heat - Worksheet D-2

On Worksheet D-2, first select an area of solar collector for study, based on experience, similar design, or arbitrary size (a collector area approximately one-fourth to one-half the floor area to be heated is a reasonable guess). Area is multiplied by F_I and F_L factors from Worksheet D-1 and the product is entered on Worksheet D-2. Then from Figure 3-1 pick off the values of f for each set of values of A_cF_I and A_cF_L. Calculate average $\bar{f} = \Sigma Q_L f / \Sigma Q_L$. Then select other collector areas, larger or smaller, and repeat above procedure so that a trend may be observed in the following cost analysis calculations. Usually very sunny areas (I > 500 L/day) will have highest cost effectiveness at about $\bar{f} \sim 0.75$ and not so sunny areas (I ~ 300) at $\bar{f} \sim 0.50$.

Use of the simplified slope factor method (Figure 3-2) results in a small error in $\bar{f}$, the fraction of building heat load supplied by solar (average over 1 year). A correction curve, Figure 3-3, will account for this. The error arises because solar insolation is composed of both beam and diffuse radiation. The simplified slope factor method treats all radiation as if it were beam. The correction curve was derived by comparing the $\bar{f}$ calculated from Klein, Beckman, and Duffie (1976) slope factor method (taken as correct, but involving complex calculations) with $\bar{f}$ calculated from the simplified method. The $\bar{f}$ calculated by the simplified method will always be "too high," thus overestimating the amount of heat which could be collected from a given collector area. Thus the percent correction should be subtracted from the $\bar{f}$ by the formula:

$$\bar{f}\ \text{correct} = \bar{f}\ \text{slope factor method}\ (1-\text{error})$$

An error of about 6% will occur for an $\bar{f}$ of 0.7 if this is not corrected by Figure 3-3. Henceforth all references to $\bar{f}$ in this document will mean the corrected version as given in Figure 3-3.

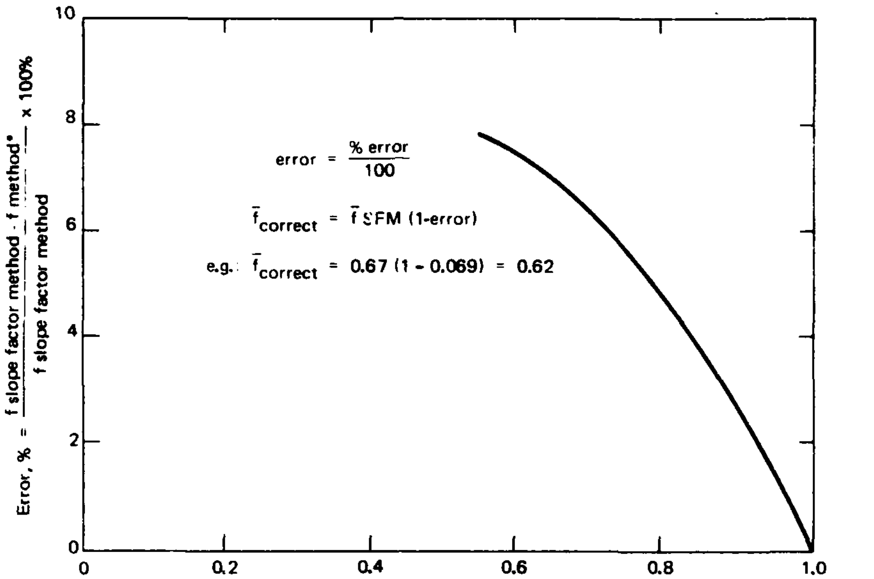

Figure 3-3. Correction curve for slope factor method.

Storage volume on Worksheet D-2 may be sized by rules of thumb for minimum size. Minimum storage volumes are: one day's usage for DHW only and 1.8 gal/ft^2 (collector) for space heating and DHW. (See Section 2.2 for sizing storage.) Up to 2.5 days usage for DHW only has been recommended for family dwellings without auxiliary heat for DHW. Up to 5 gal/ft^2 has been used in some installations for space heating and DHW. The parametric chart (Figure 3-1) has been developed using storage equal to 1.8 gal of water/ft^2. Results will not be greatly affected by moderate deviations from this value. For liquids other than water the storage figures are modified by multiplying by the ratio of specific heats: C_p water/C_p liquid = 1.0/C_p liquid. Minimum volume of rocks for air system storage is 0.8 ft^3 per square foot of collector (Section 2.2). If multiple cloudy days are a frequent occurrence, then more auxiliary heat will be used than was planned; the latter problem is relieved if larger storage is used. Consequently, if many cloudy days are expected, then the high end of the "rules of thumb" for storage sizing should be selected (Section 2.2). The cost

of energy storage may be calculated from Table 2-8, where, for the approximate size chosen, the various elements of tank order-of-magnitude cost are listed in terms of \$/gal. For instance, for an installed, 100 psi, lined steel tank, add the costs of unlined pressure tank, tank liner and installation. This figure is entered on Worksheet F. If the ultimate result of the analysis, Worksheet A, shows a cost effective system, then storage size can be increased from the minimum. Increased storage size saves fuel, and reduces the uncertainty in meeting the predicted $\bar{f}$ due to the approximate averaged method used here to calculate the heating load.

3.7 Fuel Savings - Worksheets E-1 and E-2

On Worksheet E-1, the value of the fuel saved by the solar heat collected is calculated for the several collector areas chosen.

$$\text{Value of fuel} = \frac{\bar{f} \times Q_{L_t} \times C_f}{\eta_w}$$

where $\bar{f}$ = yearly average fraction of heat load supplied by solar heat. Use $\bar{f}_{corrected}$ from Worksheet D-2.

Q_{L_t} = total yearly heating and DHW load (10^6 Btu)

C_F = cost of fuel in \$/10^6 Btu

η_w = utilization efficiency of space heater, DHW heater or an average efficiency in combined system

For the purpose of calculating \$/10^6 Btu, approximately:

Gas, one therm	=	10^5 Btu
1000 ft^3	=	10^6 Btu
LPG, propane, butane	=	95,500 Btu/gal
Bituminous coal	=	2.46×10^7 Btu/short ton
Purchased steam	=	1,390 Btu/lb

Oil, No. 2 = 140,000 Btu/gal: 5.84×10^6 Btu/bbl

No. 5 = 150,000 Btu/gal: 6.3×10^6 Btu/bbl

Electricity (at source which is fossil fueled) = 11,600 Btu/kwh

1 kwh = 3,413 Btu

The present worth of fuel saved can be calculated many different ways. Two methods are shown in the following two sections.

3.7.1 Present Worth Using A Discount Factor and Fuel Differential Escalation Rates. Historically this has been the method used and was presented in the original version of this report. It is maintained here for continuity and in the event it may be used again. The present worth of 25 years fuel saved may be calculated by using Worksheet E-2, or if standard fuel differential inflation rates (expected percentage increase per year in fuel prices above general inflation rate) and 10% discount factor are used, then a single multiplication given in note (2), Worksheet E-1 will give the answer. Annual fuel inflation factors for use on Worksheet E-1 or E-2 are given in Table 3-5a. Typically differential inflation factors of at least 7%-9% are used for fuels. If this method is used, NAVFAC INSTRUCTIONS will provide the correct rates. The figures from either Worksheet E-1, note 2 or from Worksheet E-2 are transferred to Worksheet A, column (y). Complete for each collector area under study.

3.7.2 Present Worth Using Fuel Escalation Rates Without the Discount Factor. This is the current method to be used per NAVFACINST 11010.14M, 14 Dec 78. Solar systems shall be targeted to provide a minimum of 25% of space heating load and 35% of DHW heating load. Any system will be considered economic where the initial investment cost is recovered in energy savings over the life of the facility (25 years). Maintenance and replacement costs will not be considered in the analysis. The discount factor shall not be applied. The following escalation rates shall be used:

Table 3-5a. Annual Fuel Inflation Factors, 10% Discount

[Compound Amount Factors – Factor B, Worksheet E-2, or Column (2), Worksheet E-1]

Example Year	0% Inflation 10% Discount		6% Inflation 10% Discount		7% Inflation 10% Discount		8% Inflation 10% Discount		9% Inflation 10% Discount		10% Inflation 10% Discount	
	Single Amount	Cumulative Series	Single Amount	Cumulative Series	Single Amount	Cumulative Series	Single Amount	Cumulative Series	Single Amount	Cumulative Series	Single Amount	Cumulative Series
1	0.954	0.954	0.982	0.982	0.986	0.986	0.991	0.991	0.995	0.995	1.000	1.000
2	0.867	1.821	0.946	1.928	0.959	1.946	0.973	1.964	0.986	1.982	1.000	2.000
3	0.788	2.609	0.912	2.839	0.933	2.879	0.955	2.919	0.977	2.959	1.000	3.000
4	0.717	3.326	0.878	3.718	0.908	3.787	0.938	3.857	0.969	3.928	1.000	4.000
5	0.652	3.977	0.847	4.564	0.883	4.670	0.921	4.777	0.960	4.887	1.000	5.000
6	0.592	4.570	0.816	5.380	0.859	5.529	0.904	5.681	0.951	5.839	1.000	6.000
7	0.538	5.108	0.786	6.166	0.836	6.364	0.888	6.569	0.942	6.781	1.000	7.000
8	0.489	5.597	0.757	6.923	0.813	7.177	0.871	7.440	0.934	7.715	1.000	8.000
9	0.445	6.042	0.730	7.653	0.791	7.968	0.856	8.296	0.925	8.640	1.000	9.000
10	0.405	6.447	0.703	8.357	0.769	8.737	0.840	9.136	0.917	9.557	1.000	10.000
11	0.368	6.815	0.678	9.035	0.748	9.485	0.825	9.961	0.909	10.465	1.000	11.000
12	0.334	7.149	0.653	9.688	0.728	10.212	0.810	10.770	0.900	11.366	1.000	12.000
13	0.304	7.453	0.629	10.317	0.708	10.920	0.795	11.565	0.892	12.258	1.000	13.000
14	0.276	7.729	0.607	10.924	0.688	11.608	0.781	12.346	0.884	13.142	1.000	14.000
15	0.251	7.980	0.584	11.508	0.670	12.278	0.766	13.112	0.876	14.018	1.000	15.000
16	0.228	8.209	0.560	12.071	0.651	12.930	0.752	13.865	0.868	14.886	1.000	16.000
17	0.208	8.416	0.543	12.614	0.634	13.563	0.789	14.603	0.860	15.746	1.000	17.000
18	0.189	8.605	0.523	13.137	0.616	14.180	0.725	15.329	0.852	16.598	1.000	18.000
19	0.172	8.777	0.504	13.641	0.600	14.779	0.712	16.041	0.845	17.443	1.000	19.000
20	0.156	8.933	0.486	14.127	0.583	15.363	0.699	16.740	0.837	18.279	1.000	20.000
21	0.142	9.074	0.468	14.595	0.567	15.930	0.687	17.427	0.829	19.109	1.000	21.000
22	0.129	9.203	0.451	15.046	0.552	16.482	0.674	18.101	0.822	19.930	1.000	22.000
23	0.117	9.320	0.435	15.480	0.537	17.019	0.662	18.762	0.814	20.745	1.000	23.000
24	0.107	9.427	0.419	15.899	0.522	17.541	0.650	19.412	0.807	21.551	1.000	24.000
25	0.097	9.524	0.404	16.303	0.508	18.049	0.638	20.050	0.800	22.351	1.000	25.000

NOTES: 1. Consult NAVFAC INSTR for latest fuel inflation factors.
Consult NAVFAC Manual P-442 for Compound Amount Factor tables for inflation rates not given here.
2. These fuel inflation factors are applied to costs which are anticipated to escalate at a rate i% faster than general price levels, where i is the fuel inflation factor.

	FY79 thru FY80	FY81 thru FY83	FY84 and Beyond
Fuel Oil	16%	14%	8%
Natural Gas & LPG	15%	14%	8%
Electricity	16%	13%	7%

The fuel inflation factors for this method are given in Table 3-5b. Note that depending on when a project is initiated various factors will apply. The present worth can be calculated in several steps as follows:

(1) Calculate fuel saved per year from (1), Worksheet E-1 = A

(2) Project starts in FY80 = year 1

(3) Then next 3 years (FY81-FY83) savings = 4.992A (from Table 3-5b, 14% (fuel oil), year 4 amount)

(4) The remaining 21 years of economic life are at 8% escalation (fuel oil, FY84 and beyond), therefore savings = 50.423A (Table 3-5b, 8% (fuel oil), year 21 amount)

(5) Total savings = 50.423A + 4.992A = 55.415A or 55.4 times the amount calculated in Worksheet E-1, note (1)

(6) These calculations can be done on Worksheet E-1, col (4) and then transferred to Worksheet A, column (y)

If the reader uses different economic techniques at his duty station (ECIP projects, etc.) then these worksheets need not be used. Nevertheless, the methods proposed will provide a valid comparison of solar systems and give a reasonable measure of their cost effectiveness.

3.8 Collector Temperatures - Worksheet E-1

Fluid temperature rise through collector is of interest and may be calculated from:

Table 3-5b. Fuel Inflation Factors - No Discount

[Annuity Amount Factors - Column (4), Worksheet E-1]

Year	Escalation Rates							
	6%	7%	8%	10%	13%	14%	15%	16%
1	1.000	1.000	1.000	1.000	1.000	1.000	1.000	1.000
2	2.060	2.070	2.080	2.10	2.13	2.14	2.15	2.16
3	3.184	3.2149	3.246	3.310	3.407	3.440	3.473	3.506
4	4.375	4.410	4.506	4.641	4.850	4.922	4.994	5.067
5	5.637	5.751	5.867	6.105	6.480	6.611	6.743	6.878
6	6.975	7.153	7.336					
7	8.394	8.654	8.923					
8	9.897	10.260	10.637					
9	11.491	11.978	12.488					
10	13.181	13.816	14.486					
11	14.972	15.784	16.645					
12	16.870	17.888	18.977					
13	18.882	20.141	21.495					
14	21.015	22.550	24.215					
15	23.276	25.129	27.152					
16	25.673	27.888	30.324					
17	28.213	30.840	33.750					
18	30.906	33.999	37.450					
19	33.760	37.379	41.446					
20	36.786	40.995	45.762					
21	39.993	44.865	50.423					
22	43.392	49.006	55.457					
23	46.995	53.436	60.893					
24	50.815	58.176	66.765					
25	54.865	63.249	73.106					

NOTE: Table gives amount accumulated at end of n years by a given annual payment (annuity) (A, Sec. 3.7). Factors are for escalation rates compounded annually.

$$T_o - T_i = \frac{f\ Q_L/N_o}{C_p\ \theta\ G}$$

where f = fraction supplied by solar (Worksheet D-2 for month selected)

T_o = collector outlet temperature

T_i = collector inlet temperature

G = flowrate in lbm/hr ft^2 collector (use 10.0 lbm/hr ft^2)

Q_L = heat load, Btu/mo (Worksheet C-1)

C_p = specific heat of fluid = 1 Btu/lbm F for water

θ = hours of useful sun in day (use 5 or 6 hours)

If a DHW only system is used, then temperature rise may be added to ground water temperature to obtain actual collector outlet temperature. For space heating/DHW systems, the minimum collector outlet temperatures will be the temperature of the water returning from the room heat exchanger plus the temperature rise through the collector. In general, the storage tank bottom temperature is added to temperature rise to obtain actual collector outlet temperature.

3.9 Solar System Cost - Worksheet F

Worksheets F and G may be used to convert all costs of the solar installation into cost/ft^2 collector. Since costs can differ significantly for space heating/DHW compared to DHW only, two separate columns are shown. Recent manufacturers' data are best for computations, but Tables 3-6 and 3-7 may be used as representative prices (based on data as of December 1978). Contractor profit is indicated as 20% and is included in tables; another figure may be used if warranted. Solar collector costs are also given in Table 2-6. Total system cost estimate is transferred to Worksheet A.

Table 3-6. Solar System Component Cost Estimates[a]

Item	Space/DHW ($)	DHW ($)
Antifreeze	$0.10/ft^2$	$0.10/ft^2$
Pumps, pipe, controls	$4.25/ft^2$	$6.00/ft^2$
• Heat exchanger	$1.50/ft^2$	$1.50/ft^2$
Auxiliary heater		
• Gross amount	$6.25/ft^2$	$1.00/ft^2$
• Less value of conventional system	$(4.00/ft^2)$	$(1.00/ft^2)$
Auxiliary heater, net	$2.25/ft^2$ net	$0/ft^2$ net

[a]December 1978.

Table 3-7. Solar Collector Prices[a]

Type	Unit Selling Price ($\$/ft^2$)
Plastic, no cover	5.00
Aluminum and copper single g[illegible]ze	8.00-15.00
Copper, double glazed	18.00-20.00
Aluminum, single glaze free flow (trickle)	5.00-8.00
Plastic, single glaze with insulation	7.00-9.00

[a]December 1978.

NOTES:
1. See Table 2-6 for more specific comparison of cost, collector type, performance, etc.
2. Installed cost is usually double (80%-100%) the selling price.

3.10 Additional Costs - Worksheet G

Worksheet G is a convenient checklist to collect costs associated with converting to solar energy. On new building designs, good insulation, weatherstripping, etc., will be called for to save energy, even if solar heating is not adopted, thus the solar system should not be burdened with these costs in new buildings. Costs are summed and divided by collector area, then cost is transferred to Worksheet F. Maintenance cost can vary from 1% of total systems cost in large installations to 5% in single residence DHW applications.

3.11 Sizing the Heat Exchanger for Space Heating

According to Klein, Beckman, and Duffie (1976):

> The dimensionless parameter $\varepsilon_L C_{min}/UA$, has been found to provide a measure of the si?e heat exchanger needed to supply solar heat to a specified building. For values of $\varepsilon_L C_{min}/UA$ less than 1.0, the reduction in system performance due to too small a heat exchanger will be appreciable. Reasonable values of $\varepsilon_L C_{min}/UA$ for solar space heating systems are between 1 and 3 when costs are considered. (This design method has been developed) with $\varepsilon_L C_{min}/UA$ equal to 2.0.

C_{min} is heat capacity flowrate, which is the lesser of the two heat capacity flowrates in the load heat exchanger; ε_L is effectiveness of load heat exchanger and UA is overall heat loss coefficient of building times the building area.

3.12 Air-Heating Collector Design - Worksheet H

The design procedure for air systems (Klein, Beckman, and Duffie, 1977) is very similar to that for liquid systems - the same worksheets may be used. Figure 3-4 gives the f-chart for this procedure. The procedure was developed using an air flow heat capacitance

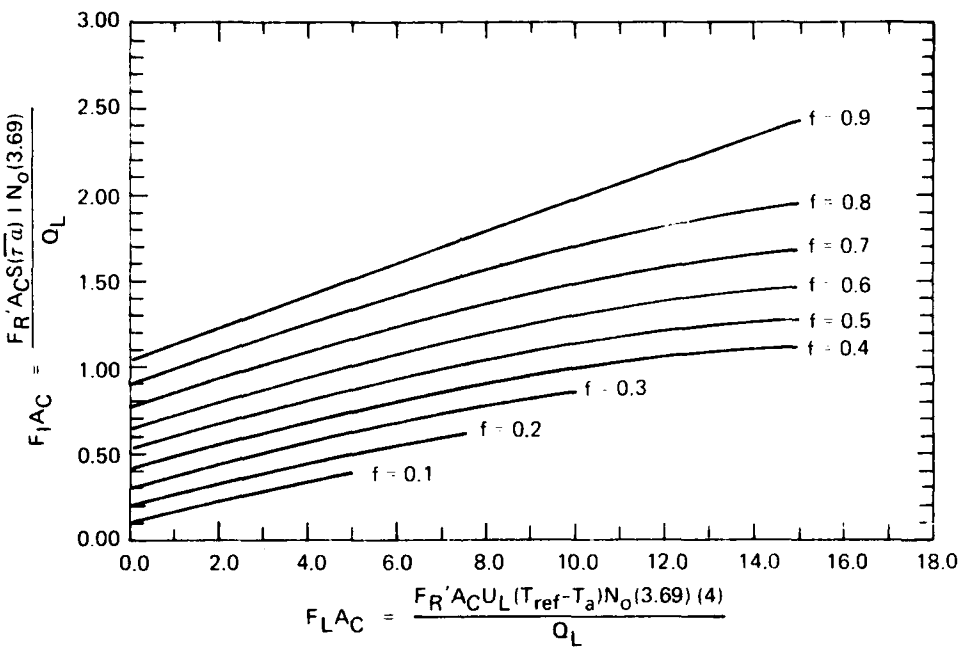

Figure 3-4. Fraction of space heating/DHW load supplied by solar air heating system (after Klein, Beckman, and Duffie, 1977).

rate of 2.87 Btu/hr ft^2 °F (about 156 SCF/hr ft^2). The performance of systems having collector capacitance rates between 1.47 and 5.87 Btu/hr ft^2 F can be estimated by multiplying the values of $F_L A_c\{[(\dot{m}c_p)_c/F_R A_c]/2.87\}^{0.28}$ (Klein, Beckman, and Duffie, 1977). To calculate F_R see Duffie and Beckman (1974), Section 7.7. The rock bed storage heat capacitance assumed was 19.6 Btu/ft^2 F. The performance of systems with other storage capacities can be determined by multiplying the dimensionless group $F_L A_c$ by $[(\rho V C_{pr}/F_R A_c)/19.6]^{-0.3}$ (Klein, Beckman, and Duffie, 1977). The standard deviation of the yearly $\bar{f}$ by this method from the computer simulated value was found to be 0.017 (Klein, Beckman, and Duffie, 1977).

> A comparison of the f-charts for the liquid and air systems indicates that, for the same values of $[F_I A_c]$ and $[F_L A_c]$ the air system outperforms the liquid system particularly for systems designed to supply a large fraction of the heating load....[Some reasons for this behavior are:] the average collector fluid inlet temperature is lower for the air system (and thus the collector efficiency is higher) than that for the liquid system at times when . . . room air is circulated through the collector and also because the more effective thermal stratification achieved in rock storage results in lower temperature air going to the collector. Also since no heat exchanger is required between collector and storage, that inefficiency is avoided.
>
> It cannot be concluded, however, that air heating systems perform better than liquid systems. The overall collector efficiency factor, F_R, is ordinarily lower for air heaters. As a result $[F_I A_c]$ and $[F_L A_c]$ are ordinarily lower and thus the performance of an air system may be equivalent to or lower than that of a liquid system, all else being the same. (Klein, Beckman, and Duffie, 1977).

Sections 2.1.1 and 2.5 of this document discuss the relative merits of air systems.

3.13 Worksheets

A	Job Summary
B	Solar Collector Parameters
C-1	Load Calculations
C-2	Demand Calculations - Domestic Water Heater
D-1	Monthly Solar Collection Parameters
D-2	Fraction of Load Supplied by Solar Heat
E-1	Value of Fuel Saved
E-2	Present Worth Analysis
F	Solar System Cost Analysis
G	Additional Cost Items Related to Use of Solar Heating
H	Solar Air Collector System Design Summary

WORKSHEET A

Job Summary

Date ____________ Job Number ____________________

Building ____________________ General Construction ____________________

Location ____

Occupancy ____________________ Hours of Occupancy ____________________

Type of Solar System ____

Building Area ____________ ft^2 No. BR ____________________ No. Baths ____________

Fuel ________ Burned in ____________ @ ______ % assumed @ Cost ________ $/10^6$ Btu
efficiency

________ ____________ @ ______ % ________ $/10^6$ Btu

________ ____________ @ ______ % ________ $/10^6$ Btu

Solar Collector Description ____

Approx. Cost, installed, total system $__________ $/ft^2$ D.H.W.
(Worksheet F)

$__________ $/ft^2$ Space Heating/D.H.W.

(1)		(x)	(y)		
Area (From E-1)	$\overline{f}$ (From E-1)	Solar System Cost (From F)	Value of 25 yr. Fuel Saved (From E-1 or E-2)	Savings/ Investment Ratio (y)/(x)	Payback Period (Table C NAVFAC Manual P 442)

(1) From Worksheet E-1.

WORKSHEET B

SOLAR COLLECTOR PARAMETERS

JOB NO. ________________

(1) $F_R(\tau\alpha)_n$ = __________

(2) $F_R U_L$ = __________ Langleys/($^\circ$F day)

(3) $(\dot{m}C_p)_c/A_c$ = __________

(4) ϵ_c = __________

(5) $\dfrac{(\dot{m}C_p)_c}{(\dot{m}C_p)_{min}}$ = __________

(6) $\dfrac{F_R'}{F_R} = \left\{ 1 + \left[F_R U_L \left(\dfrac{A_c}{(\dot{m}C_p)_c}\right)\right] \left[\dfrac{(\dot{m}C_p)_c}{\epsilon_c(\dot{m}C_p)_{min}} - 1\right] \right\}^{-1}$ = __________

(7) $\dfrac{\overline{(\tau\alpha)}}{(\tau\alpha)_n}$ = ___

$F_R'\,\overline{(\tau\alpha)} = \left(\dfrac{F_R'}{F_R}\right)\left(\dfrac{\overline{(\tau\alpha)}}{(\tau\alpha)_n}\right) F_R(\tau\alpha)_n$ = __________

$F_R'U_L = \left(\dfrac{F_R'}{F_R}\right) F_R U_L$ = __________

(1) Obtained from y-intercept of η vs $\dfrac{\Delta T}{I}$ curve, Table 2-6, or manufacturer's data.

(2) Obtained from absolute value of slope of η vs $\dfrac{\Delta T}{I}$ curve, Table 2-6. Units of Langleys/($^\circ$F day) are for a 6-hour sunlight day, not a 24-hour day. This must be observed when other units such as Btu/ft^2 $^\circ$F hr are used.

(3) Mass flowrate of working fluid through collector, $\dot{m}$; specific heat of fluid C_p; area of collector, A_c. May use constant = 10 Btu/hr ft^2 F.

(4) Effectiveness of the collector-tank heat exchanger, if employed; if not employed, use ϵ_c = 1.0.

(5) Ratio of heat capacity flowrate of the fluid through the collector to the heat capacity flowrate which is the minimum of the two fluids in the collector-tank heat exchanger, if employed; if not employed, use ratio = 1.0.

(6) Will equal 1.0 if no collector-tank heat exchanger employed.

(7) Use constant if no better data available. Constant = 0.91 for double glazed collector; 0.93 for single glazed collector

WORKSHEET C-1

LOAD CALCULATIONS (5)

JOB NO. ____________________

Heat Loss Rate (L) ____________________ B/ft^2 degree-day gross (from Table 3-1)
or net

Area (M) ____________________ ft^2

Year 19 _____

Month	Degree Days (P)	GROSS		NET		
		Space Heat Load R=(L)x(M)x(P)	Hot Water (U)	Space Heat Load (V)=(R x ηw)	Hot Water (W) $Q_d \times N_o$	Total Q_l = (V)+(W)
DEC						
JAN						
FEB						
MAR						
APR						
MAY						
JUN						
JUL						
AUG						
SEP						
OCT						
NOV						
	(1)	(2)	(3)	(4)	(5)	
					$\sum_{12} Q_l = Q_{l,t}$	

(1) From local records or Climatic Atlas of U.S., U.S. Dept. Commerce. **Excerpts in Table 3-2.**

(2) Based on fuel used

(3) From Worksheet C-2, Gross = $\frac{\text{net}}{\eta_w}$, η_w = utilization efficiency of heater. May be approximated as constant.

(4) η_w = Utilization efficiency of heater. Net space heat may be calculated from heat loss of building or from fuel usage times efficiency of heater. If "L" is net heat loss rate, then "V" = LxMxP (without η_w).

(5) Units of heat on this Worksheet are in 10^6 Btu

WORKSHEET C-2

DEMAND CALCULATIONS DOMESTIC WATER HEATER

JOB NO ________________

Type Building ____ BR ______ Bath __________

No. of Occupants ________________________________ Use/day-person(1) ______________

Average daily demand, gallons ______ x 8.3 lbs./gal. = ____________________ lbs W

Supply temperature (winter), ^{o}F ________ (2) Average water temperature (T_i)

After heating ________________ ^{o}F – Desired hot water temperature (T_o)

Q_d = daily BTU's to be collected = $W\ C_p\ \Delta T = W\ C_p\ (T_o - T_i)$

______ lb. (1.0) ______ ^{o}F ________________ B day

Month	(2) T_i ^{o}F	(3) Q_d BTU's required one day	N_o No. of days in month	Net Monthly Average Demand $Q_d \times N_o$
DEC				
JAN				
FEB				
MAR				
APR				
MAY				
JUN				
JUL				
AUG				
SEP				
OCT				
NOV				
			$\Sigma Q_d N_o = Q_{d_t}$	

(1) Source – Table 2-9 (Section 2.3) or DM-3, Chapter 1.

(2) Ground water temperature taken as normal daily average temperature, Table 3-3 or from Climatic Atlas of US, US Department of Commerce (US NOAA, 1968)

(3) May be approximated as constant, or accuracy may be improved by using different T_i and T_o for each month.

WORKSHEET D-1

MONTHLY SOLAR COLLECTION PARAMETERS

JOB NO ______________________

$F_R' \overline{(\tau\alpha)}$ = ________________ (from Worksheet B)

$F_R' U_L$ = ________________ (from Worksheet B)

Mo.	(3) N_o (days/mo.)	(3) I (lys/day)	(4) S Slope Factor	(6) Air Temp T_a (^oF)	$T_{ref}-T_a$ = (212F-Ta) (^oF)	(1) Q_L (10^6B/mo.)	(1,2) $F_1 = \frac{N_o F_R' \overline{(\tau\alpha)} IS(3.69)}{Q_L}$ (ft^{-2})	(1,2,5) $F_2 = \frac{F_R' U_L (T_{ref}-T_a) N_o (3.69)(4.0)}{Q_L}$ (ft^{-2})
DEC								
JAN								
FEB								
MAR								
APR								
MAY								
JUN								
JUL								
AUG								
SEP								
OCT								
NOV								

(1) Loads, Q_L, from Worksheet C-1

(2) Factor 3.69 converts langleys/day to BTU/ft^2 day.

(3) From Table 1-1 based on location: ______________________

(4) From Figure 3-2 based on tilt angle of latitude ________ + 10° = __________

(5) Factor (4.0) converts hours of sunlight (6 hours) to hours per day (24 hours)

(6) Table 3-4 or US NOAA, 1968, Climatic Atlas of the United States.

WORKSHEET D-2

FRACTION OF LOAD SUPPLIED BY SOLAR HEAT

JOB NO. ____________

Month	A_c = ______ ft^2			A_c = ______ ft^2			A_c = ______ ft^2		
	$A_c F_I$ (1)	$A_c F_L$ (1)	f (2)	$A_c F_I$	$A_c F_L$	f (2)	$A_c F_I$	$A_c F_L$	f (2)
DEC									
JAN									
FEB									
MAR									
APR									
MAY									
JUN									
JUL									
AUG									
SEP									
OCT									
NOV									
		$\bar{f} = \frac{\Sigma Q_L f}{\Sigma Q_L}$							
				(3) $\bar{f}_{corrected}$					

Note: use Q_L's from Worksheet D-1.

STORAGE SIZING:

Minimum storage size - DHW one days' usage (Worksheet C-2) V = ________ gal.

Space heat/DHW 1.8 gal/ft^2 collector (Section 2.2) V = 1.8 x A_c = ________ gal.

For non-water, see Section 3.6

Other "rules of thumb" –

DHW 1.0 - 2.5 day's usage (the latter with no auxiliary heater) V = ________ gal.

Space heat/DHW: 1.8 - 5 gal/ft^2 (see Sections 2.2, 3.6) V = ______ x A_c ______ gal.

(1) F_I and F_L from Worksheet D-1

(2) From Figure 3-1 after A_cF_I and A_cF_L calculated

(3) See Figure 3-3, Section 3.6

WORKSHEET E-1

VALUE OF FUEL SAVED

JOB NO. ______________

(3)					
A_c Area From Worksheet D-2 (ft^2)	$\bar{f}$ Fraction Supplied by Solar	$Q_{L\text{-}t}$ (From Worksheet C-1) (BTU x 10^6/yr)	(1) Value of Fuel Saved per Year A	(2) Present Worth of 25 yrs. Fuel at __%/yr. Inflation Rate and 10% Discount Rate, Table 3-5a	(4) Present Worth of 25 yrs. Fuel at __%/yr. Inflation Rate No Discount, Table 3-5b

(1) A = Value of fuel saved = $\dfrac{\bar{f} \times Q_{L\text{-}t} \times C_F}{\eta_w}$, use $\bar{f}_{corrected}$ from Worksheet D-2

C_F = cost of fuel, \$/10^6 BTU

η_w = Utilization efficiency of furnace, DHW heater or compromise η_w. (Worksheet A)

(2) Present Worth = 18.049 x value of fuel saved/yr - 7% Inflation
20.050 x value of fuel saved/yr - 8% Inflation
25.000 x value of fuel saved/yr - 10% Inflation
} 10% Discount See Table 3-5a, Section 3.7.1

or use Worksheet E-2. Note: Consult NAVFAC Instr for latest fuel inflation rates.
Transfer figures to Worksheet A.

(3) Or Q_{d_t} from Worksheet C-2 for DHW only systems.

(4) No discount method (see Section 3.7), Table 3-5b, use either column (2) or (4), not both.
Transfer figures to Worksheet A.

COLLECTOR TEMPERATURES

$$T_o - T_i = \frac{fQ_L/N_0}{GC_p \theta A_c} = \underline{\qquad\qquad\qquad}$$

Q_L from C-1 for month selected

N_0 number of days in month

C_p specific heat of fluid

θ number of hours of usable sun in day, use 5 or 6 hours

G is flowrate in lbm/ft^2hr, use (10 $lbm/hr\ ft^2$)

f from D-2 for month selected.

WORKSHEET E-2
PRESENT WORTH ANALYSIS

JOB NO. ______________

Collector Area__________ ft²

		A	B	C
Year____ (Specify CY or FY)	Year of Analysis	Dollar Value of Fuel Saved in Zeroth Year (Worksheet E-1)**	Fuel Inflation Factor at____% Discount at 10% From Table 3-5a	Annual Present Worth (A x B)
	Zeroth*			
	1st			
	2nd			
	3rd			
	4th			
	5th			
	6th			
	7th			
	8th			
	9th			
	10th			
	11th			
	12th			
	13th			
	14th			
	15th			
	16th			
	17th			
	18th			
	19th			
	20th			
	21st			
	22nd			
	23rd			
	24th			
	25th			

Present worth of fuel saved by system (transfer to Worksheet A, Column Y) Σ ______________

* Year for which fuel costs are available if year of construction. Otherwise, escalate fuel costs to year of construction.

** Copy zeroth year fuel saved into each space in Column A.

WORKSHEET E

SOLAR SYSTEM COST ANALYSIS

JOB NO. ______________________

	Item Description	Area ______ ft^2 Space Heating/DHW	______ ft^2 DHW Only
(1)	Collector	/ft^2	/ft^2
(2)	Storage tank, installed, ____________ \$______ /gal x ______ gal/ft^2 collector	/ft^2	/ft^2
(3)	Auxiliary heating unit, installed cost/ft^2 collector, net of conventional unit cost	/ft^2	/ft^2
(3)	Pumps, pipe, heat exchangers, controls cost/ft^2 collector	/ft^2	/ft^2
	Other (from Worksheet G)	/ft^2	/ft^2
	Subtotal	______	______
(1)	Collector installation	/ft^2	/ft^2
(4)	TOTAL	______	______

(4)

Area	Cost/ft^2	Cost

(1) Manufacturer's data, or from Table 2-6 or Table 3-7, plus 80 to 100% for installation

(2) See Table 2-8 and Worksheet D-2, plus \$.10 (antifreeze) and \$1.50/ft^2 (heat exchanger) if applicable.

(3) See Table 3-6.

(4) Transfer totals to Worksheet A.

WORKSHEET G

ADDITIONAL COST ITEMS RELATED TO USE OF SOLAR HEATING

JOB NO. ____________________

COST ITEM (Capital costs this sheet)	ATTRIBUTED TO PLANNED SOLAR SYSTEM		
	Yes	No	Cost
Change or add unit heaters			
Change or add circulating pumps			
Change or add controls, e.g., to radiators, attic exhaust fan			
Increase in interior floor space to accommodate tempering or storage tanks, pumps, etc.			
Excavation and backfill, storage tank			
Elimination of excess standby boilers, furnaces, etc.			
Capital value of space obtained by eliminating boilers, etc. in above item.			
Electricity for pumps, fans - excess cost over conventional system			
Maintenance @ 1%-5% of total system cost/year.			
Other			
Total			

Convert to $\$/ft^2$ collector:

A_c = __________ ft^2

Total = __________ $\$/ft^2$, transfer to Worksheet F.

WORKSHEET II

SOLAR AIR COLLECTOR SYSTEM DESIGN SUMMARY

JOB NO. ______________________

			(1)	(3)	(2)	(2)
Month	Load Q_L B/mo. (from C-1)	Solar q_c B/mo. ft^2 (see Sec. 3.12)	Area - ft^2 $= \frac{Q_L}{q_c}$	USEFUL HEAT PRODUCED $Q_u = A_c q_c$ B/mo. A_c = ___	$Q_u = A_c q_c$ B/mo. A_c = ___	$Q_u = A_c q_c$ B/mo. A_c = ___
DEC						
JAN						
FEB						
MAR						
APR						
MAY						
JUN						
JUL						
AUG						
SEP						
OCT						
NOV						
Total Year			Totals/yr.			
			$\bar{f} = (\Sigma Q_u)/Q_{L_t}$			
				(3)	(3)	(3)

(1) Area $= \frac{Q_L}{q_c}$ where Q_L = heating load; q_c = solar heat collected/ft^2 collector.

(2) Subsequent columns of useful heat produced by lesser areas are provided to allow determination of value of lesser areas, if maximum cannot be justified.

*(3) In assigning values of useful heat, to the right of the double line, no more can be credited to the system for heat saving than the load can use. Enter the lesser of the two values, required or collected, for a given area. When Q_L is entered, identify the value with an asterisk (*).

4.0 EXAMPLE PROBLEMS

See Section 3 for instructions on preparing Worksheets.

4.1 Discussion of Example 1

Space and Water Heating System for Family Housing (see Section 4.1.1). Preliminary job data are entered on Worksheet A. The first step in the analysis (Worksheet B) is to determine the y-intercept and slope of the collector efficiency curve. A single glazed, all copper rollbond collector with selective coating was chosen. The y-intercept and slope were taken directly from Table 2-6, Collector #22. They are 0.77 and 1.72 Langleys/°F/day, respectively. Note absolute value of slope used. Next, the recommended figure of 10 Btu/ft² °F hr is selected for $(\dot{m}C_p)_c/A_c$, line 3, Worksheet B. Since there will be no heat exchanger between collector and tank fluids, the next three factors equal 1.0. Line 7, Worksheet B is equal to 0.91 for a single glazed collector. Then $F_R'(\tau\alpha)$ and $F_R'U_L$ are calculated and transferred to Worksheet D-1.

The next step is to determine the heat load or demand. This is usually done by conventional methods of estimating heat losses from buildings and water usage per occupant. Table 3-1 provides estimates of building heat loss rates and other sources for calculating this parameter are given in Section 3.3. In Example 1, Worksheet C-1, the fuel usage was calculated using the Btu/ft² degree-day method. First a rough estimate for the average family house at Port Hueneme was obtained from 1 month's usage of gas for all housing divided by the number of degree days in the month and the total number of square feet in the housing area. This figure (Q_L = 29.Btu/ft²dd) included hot water heating. The estimated hot water use for a 3-bedroom 2-bath home gave a figure for hot water use per square foot which was then subtracted from total use for the 1,500 ft² home. Resulting figure gave space heating fuel use as 21.5 Btu/ft²dd. This gross figure was multiplied by furnace efficiency of 0.7 to get 15.0 Btu/ft²dd net. Net heat is that which must be supplied by solar heat. Gross heat represents

the heat value of fuel used by a conventional system. Heating degree days in Worksheet C-1 are from local records. Table 3-2 can be used for other locations.

Worksheet C-2 is used to calculate DHW use. Water usage is determined from Table 2-9, Section 2.3 or other. For Example 1, water usage of 30 gal/day/person was chosen (from Section 2.3) and water main temperatures from Table 3-3, Los Angeles, were used. Worksheets C-1 and C-2 can now be completed. The DHW demand from Worksheet C-2 is transferred to Worksheet C-1 and the Q_L from Worksheet C-1 is transferred to Worksheet D-1. The DHW demand could have been approximated as a constant at the highest use for conservative design, but the calculations are made in Worksheet C-2 for illustrative purposes.

To complete Worksheet D-1, select from Table 1-1 the nearest or most meteorologically similar location (same latitude and degree of cloudiness). Enter insolation in Langleys/day and slope factors from Figure 3-2 for the appropriate latitude. Air temperature, T_a, is obtained from Table 3-4. For this example, Los Angeles was selected as the nearest similar location (latitude = 34°) and the slope factor was based on latitude plus 10° to emphasize winter heating (Section 1.3.3).

Worksheet D-2 is begun by selecting collector area of 200 ft^2 as an arbitrary size. Then 200 ft^2 was multiplied by F_I and F_L from Worksheet D-1. For each pair of points A_cF_I and A_cF_L, Figure 3-1 is entered to find f. When $\bar{f}$, Worksheet D-2, average yearly load carried by solar heating, is calculated, a result of 0.685 is obtained. Q_L and ΣQ_L are from Worksheet C-1. Then from Figure 3-3, $\bar{f}_{corrected}$ is calculated to be 0.637. Another area of 300 ft^2 is selected and found to provide 0.795 of the load. A smaller area could have been evaluated to provide the DHW load only. This information is transferred to Worksheet E-1. Storage is selected on the basis of guidelines given in Section 2.2 (1.8 gal/ft^2) to give 360 gal and 540 gal for the two collector areas investigated.

Use Worksheet E-1 to determine value of fuel saved. The present worth is calculated from columns 2 or 4 of Worksheet E-1 (see Section 3.7 for explanation of which to use). For this example, the no discount method, natural gas fuel, and appropriate inflation factors (Table

3-5b) are used. Therefore, column (2) is not used nor Worksheet E-2. Assume project starts in FY80, then compute column (4) as example shown in Section 3.7.2 (i.e., 55.415 times amount in column 1).

Fluid temperature rise through the collector is calculated on Worksheet E-1 for a selected month. Calculation may be repeated for each month desired. For the month of December and 200 ft^2 collector area, f is 0.5, Q_L is 9.31×10^6 Btu/mo, G is 10 lb/hr ft^2. Using the formula on Worksheet E-1, a temperature rise of 12.5°F is found. This is the order of magnitude temperature rise most desired. Note that, using the same formula, the ΔT could be fixed at, say 10°F, and a flowrate, G, calculated. Sizing of pump, pipe, and collector tube diameters for low pressure drop is based on flowrate G (see Section 2.9).

Worksheet F may be used to calculate total installed solar system cost/ft^2, or, the manufacturer's price for complete systems, installed, based on per square foot of collector area, may be used. On Worksheet F, collector cost may be obtained from Table 3-7, Table 2-6, or manufacturers' quote may be used. Tank volume is based on 1.8 gal/ft^2 collector for space heating/DHW. The installed price of the insulated tank was taken from Table 2-8 (here \$3.50/gal). Up to 5 gal/ft^2 may be specified for space heating/DHW. DHW-only storage may equal one or more days expected consumption. Other costs may be priced from a detailed design or figures from Table 3-6 may be used. Other costs listed on Worksheet G are neglected for this example for simplicity.

Worksheet A may now be filled in with the values from the various worksheets. When this is done the solar system using 200 ft^2 of collector is found to be cost effective in that it saves more in fuel than it costs to install. The 300 ft^2 system is not cost effective. This fact may also be seen from Worksheet D-2 for the 300 ft^2 system. There are 6 months where the fraction of load supplied by solar, f, is equal to 1.0 and some of the other months are close to 1.0. Much of this system is not being utilized; that is, it does not result in fuel savings for those months when more solar energy is available than can be used.

The next step, for this example, would be to evaluate a 250 ft^2 system to see if it is more or less cost effective than the 200 ft^2 system. Also a given system can usually be made more cost effective by installing insulation to reduce the building heat loss rate. This should always be considered.

4.1.1 Example 1 - Worksheets.

WORKSHEET A

Job Summary

Date Nov 1979 Job Number Example 1

Building Family Quarters General Construction Stucco

Location Port Hueneme, California

Occupancy Family of Four Hours of Occupancy

Type of Solar System Space and DHW

Building Area 1,467 ft^2 No. BR 3 No. Baths 2

Fuel Gas Burned in Heater @ 70 % assumed efficiency @ Cost \$2.30 /10^6 Btu

@ % 10^6 Btu

@ % 10^6 Btu

Solar Collector Description single glazed glass, copper Rollbond, selective coating

Approx. Cost, installed, total system \$ /ft^2 D.H.W
(Worksheet F)

\$40.10 /ft^2 Space Heating D.H.W

(1)		(x)	(y)		
Area (From E-1)	$\bar{f}$ (From E-1)	Solar System Cost (From F)	Value of 25 yr. Fuel Saved (From E-1 or E-2)	Savings Investment Ratio (y)/(x)	Payback Period (Table C, NAVFAC Manual P-442)
200	.637	8,020	8,242	1.03	≈ 23-24 years
300	.795	12,030	10,286	0.86	N/A

(1) From Worksheet E-1.

WORKSHEET B

SOLAR COLLECTOR PARAMETERS

JOB NO Example 1

(1) $F_R(\tau\alpha)_n$ = 0.77

(2) $F_R U_L$ = 1.72 Langleys/(°F day)

(3) $(\dot{m}C_p)_c / A_c$ 10 BTU/ft² F hr

(4) ϵ_c = 1.0

(5) $\dfrac{(\dot{m}C_p)_c}{(\dot{m}C_p)_{min}}$ 1.0

(6) $\dfrac{F_R'}{F_R} = \left\{ 1 + \left[\dfrac{F_R U_L}{} \left(\dfrac{A_c}{(\dot{m}C_p)_c}\right)\right] \left[\dfrac{(\dot{m}C_p)_c}{\epsilon_c(\dot{m}C_p)_{min}} - 1\right]\right\}^{-1}$ 1.0

(7) $\dfrac{\overline{(\tau\alpha)}}{(\tau\alpha)_n}$ = 0.91

$F_R' \overline{(\tau\alpha)} = \left(\dfrac{F_R'}{F_R}\right)\left(\dfrac{\overline{(\tau\alpha)}}{(\tau\alpha)_n}\right) F_R(\tau\alpha)_n$ 0.701

$F_R' U_L = \left(\dfrac{F_R'}{F_R}\right) F_R U_L$ 1.72

(1) Obtained from y intercept of η vs $\dfrac{\Delta T}{I}$ curve, Table 2-6, or manufacturer's data.

(2) Obtained from absolute value of slope of η vs $\dfrac{\Delta T}{I}$ curve, Table 2-6. Units of Langleys/(°F day) are for a 6-hour sunlight day, not a 24-hour day. This must be observed when other units such as Btu/ft² °F hr are used.

(3) Mass flowrate of working fluid through collector, $\dot{m}$; specific heat of fluid C_p; area of collector, A_c. May use constant = 10 Btu/hr ft² F.

(4) Effectiveness of the collector-tank heat exchanger, if employed; if not employed, use $\epsilon_c = 1.0$.

(5) Ratio of heat capacity flowrate of the fluid through the collector to the heat capacity flowrate which is the minimum of the two fluids in the collector-tank heat exchanger, if employed; if not employed, use ratio = 1.0.

(6) Will equal 1.0 if no collector-tank heat exchanger employed.

(7) Use constant if no better data available. Constant = 0.91 for double glazed collector; 0.93 for single glazed collector

WORKSHEET C-1

LOAD CALCULATIONS (5)

JOB NO. Example 1

Heat Loss Rate (L) 15.0 B/ft^2 degree-day gross (from Table 3-1) or net

Area (M) 1467 ft^2

Year: 19 79

Month	Degree Days (P)	GROSS		NET		
		Space Heat Load R=(L)x(M)x(P)	Hot Water (U)	Space Heat Load (V)=(R x η_w)	Hot Water (W) $Q_d \times N_o$	Total Q_L = (V)+(W)
DEC	318			6.99	2.32	9.31
JAN	309			6.80	2.47	9.27
FEB	322			7.09	2.23	9.32
MAR	312			6.86	2.35	9.21
APR	299			6.58	2.00	8.58
MAY	184			4.05	1.91	5.96
JUN	74			1.63	1.70	3.33
JUL	18			0.40	1.73	2.13
AUG	6			0.13	1.67	1.80
SEP	30			0.66	1.64	2.30
OCT	87			1.91	1.88	3.79
NOV	182			4.00	2.06	6.06
	(1)	(2)	(3)	(4)	(5)	
					$\sum_{12} Q_L = Q_{L_T}$	71.06

(1) From local records or Climatic Atlas of U.S., U.S. Dept. Commerce. Excerpts in Table 3-2.

(2) Based on fuel used.

(3) From Worksheet C-2, Gross = $\frac{\text{net}}{\eta_w}$, η_w = utilization efficiency of heater. May be approximated as constant.

(4) η_w = Utilization efficiency of heater. Net space heat may be calculated from heat loss of building or from fuel usage times efficiency of heater. If "L" is net heat loss rate, then "V" = LxMxP (without η_w).

(5) Units of heat on this Worksheet are in 10^6 Btu.

WORKSHEET C-2

DEMAND CALCULATIONS DOMESTIC WATER HEATER

JOB NO Example 1

Type Building Quarters BR 3 Bath 2

No. of Occupants 4 Use/day/person(1) 30

Average daily demand, gallons 120 x 8.3 lbs/gal. 996 lbs W

Supply temperature (winter), °F see below (2) Average water temperature (T_i)

After heating 130 °F = Desired hot water temperature (T_o)

Q_d = daily BTU's to be collected = $W C_p \Delta T = W C_p (T_o - T_i)$

For Jan. 996 lb. (1.0) (130-55) °F 0.75x10^5 B/day

Month	(2) T_i °F	(3) Q_d BTU's required one day	N_o No. of days in month	Net Monthly Average Demand $Q_d \times N_o$
DEC	55	0.75×10^5	31	2.32×10^6
JAN	50	0.80×10^5	31	2.47×10^6
FEB	50	0.80×10^5	28	2.23×10^6
MAR	54	0.76×10^5	31	2.35×10^6
APR	63	0.67×10^5	30	2.00×10^6
MAY	68	0.62×10^5	31	1.91×10^6
JUN	73	0.57×10^5	30	1.70×10^6
JUL	74	0.56×10^5	31	1.73×10^6
AUG	76	0.54×10^5	31	1.67×10^6
SEP	75	0.55×10^5	30	1.64×10^6
OCT	69	0.61×10^5	31	1.88×10^6
NOV	61	0.69×10^5	30	2.06×10^6
			$\Sigma Q_d N_o = Q_{d_t}$	23.96×10^6

(1) Source – Table 2-9 (Section 2.3) or DM-3, Chapter 1.

(2) Ground water temperature taken as normal daily average temperature, Table 3-3 or from Climatic Atlas of US, US Department of Commerce (US NOAA, 1968)

(3) May be approximated as constant, or accuracy may be improved by using different T_i and T_o for each month.

WORKSHEET D-1

MONTHLY SOLAR COLLECTION PARAMETERS

JOB NO. Example 1

$F_R'\overline{(\tau\alpha)}$ = 0.701 (from Worksheet B)

$F_R' U_L$ = 1.72 (from Worksheet B)

Mo.	(3) N_o (days/mo.)	(3) I (lys/day)	(4) S Slope Factor	(6) Air Temp T_a (^oF)	$T_{ref}-T_a$ = (212F-Ta) (^oF)	(1) Q_l (10^6B/mo.)	(1,2) $F_I = \frac{N_o F_R' \overline{(\tau\alpha)} IS(3.69)}{Q_L}$ (ft^{-2})	(1,2,5) $F_L = \frac{F_R' U_L (T_{ref}-T_a) N_o (3.69)(4.0)}{Q_l}$ (ft^{-2})
DEC	31	228	1.9	60	152	9.31	.003731	.01285
JAN	31	243	1.7	58	154	9.27	.003573	.01307
FEB	28	337	1.4	59	153	9.32	.003666	.01167
MAR	31	446	1.2	62	150	9.21	.004649	.01282
APR	30	518	1.1	64	148	8.58	.005153	.01314
MAY	31	517	1.0	68	144	5.96	.006956	.01901
JUN	30	594	1.0	71	141	3.33	.01384	.03225
JUL	31	645	1.0	76	136	2.13	.02428	.05025
AUG	31	579	1.1	76	136	1.80	.02837	.05946
SEP	30	505	1.2	74	138	2.30	.02045	.04570
OCT	31	365	1.4	70	142	3.79	.01081	.02949
NOV	30	277	1.7	65	147	6.06	.006030	.01847

(1) Loads, Q_L, from Worksheet C-1

(2) Factor 3.69 converts langleys/day to BTU/ft^2 day.

(3) From Table 1-1 based on location: Los Angeles

(4) From Figure 3-2 based on tilt angle of latitude 34 + 10° = 44

(5) Factor(4.0) converts hours of sunlight (6 hours) to hours per day (24 hours).

(6) *Table 3-4 or US NOAA, 1968, Climatic Atlas of the United States.*

WORKSHEET D-2

FRACTION OF LOAD SUPPLIED BY SOLAR HEAT

JOB NO. Example 1

Month	A_c = 200 ft^2			A_c = 300 ft^2			A_c = ______ ft^2		
	$A_c F_I$ (1)	$A_c F_L$ (1)	f (2)	$A_c F_I$	$A_c F_L$	f (2)	$A_c F_I$	$A_c F_I$	f (2)
DEC	.746	2.57	.50	1.12	3.9	.66			
JAN	.715	2.61	.48	1.07	3.92	.68			
FEB	.733	2.33	.52	1.10	3.50	.70			
MAR	.930	2.564	.65	1.39	3.84	.83			
APR	1.031	2.63	.69	1.55	3.94	.89			
MAY	1.391	3.80	.83	2.09	5.70	1.0			
JUN	2.768	6.45	1.0	4.15	9.67	1.0			
JUL	4.856	10.05	1.0	7.28	15.1	1.0			
AUG	5.674	11.89	1.0	8.51	17.8	1.0			
SEP	4.09	9.14	1.0	6.14	13.7	1.0			
OCT	2.162	5.90	1.0	3.24	8.8	1.0			
NOV	1.206	3.69	.74	1.80	5.5	.92			
		$\bar{f} = \frac{\Sigma Q_L f}{\Sigma Q_L}$	.685			.832			
			.637	(3) $\bar{f}_{corrected}$		.795			

Note: use Q_L's from Worksheet D-1.

STORAGE SIZING:

Minimum storage size - DHW one days' usage (Worksheet C-2) V = ________ gal.

Space heat/DHW 1.8 gal/ft^2 collector (Section 2.2) V = 1.8 x A_c = 360 gal. (200 ft^2)

For non-water, see Section 3.6 540 gal (300 ft^2)

Other "rules of thumb" –

DHW 1.0 - 2.5 day's usage (the latter with no auxiliary heater) V = ________ gal.

Space heat/DHW: 1.8 - 5 gal/ft^2 (see Sections 2.2, 3.6) V = ______ x A_c ______ gal.

(1) F_I and F_L from Worksheet D-1

(2) From Figure 3-1 after A_cF_I and A_cF_L calculated

(3) See Figure 3-3, Section 3.6

WORKSHEET E-1

VALUE OF FUEL SAVED

JOB NO. Example 1

A_c Area From Worksheet D-2 (ft^2)	$\bar{f}$ Fraction Supplied by Solar	(3) QL_t (From Worksheet C-1) (BTU x 10^6/yr)	(1) Value of Fuel Saved per Year A	(2) Present Worth of 25 yrs. Fuel at __%/yr. Inflation Rate and 10% Discount Rate, Table 3-5a	(4) Present Worth of 25 yrs. Fuel at __%/yr. Inflation Rate No Discount, Table 3-5b
200	.637	71.06	$148.73	-	$8241.87
300	.795	71.06	$185.62	-	$10,286.13

(1) A = Value of fuel saved = $\dfrac{\bar{f} \times QL_t \times C_F}{\eta_w}$, use $\bar{f}_{corrected}$ from Worksheet D-2

C_F = cost of fuel, $/$10^6$ BTU

η_w = Utilization efficiency of furnace, DHW heater or compromise η_w. (Worksheet A)

(2) Present Worth = 18.049 x value of fuel saved/yr - 7% Inflation
20.050 x value of fuel saved/yr - 8% Inflation
25.000 x value of fuel saved/yr - 10% Inflation
} 10% Discount See Table 3-5a, Section 3.7.1

or use Worksheet E-2. Note: Consult NAVFAC Instr for latest fuel inflation rates.
Transfer figures to Worksheet A.

(3) Or Q_{d_t} from Worksheet C-2 for DHW only systems.

(4) No discount method (see Section 3.7), Table 3-5b, use either column (2) or (4), not both.
Transfer figures to Worksheet A.

COLLECTOR TEMPERATURES

$$T_o - T_i = \frac{fQ_L/N_0}{GC_p \theta A_c} = \frac{(.5)\ (9.31 \times 10^6)/31}{(10\ \text{lbm/hr ft}^2)\ (1\ \text{BTU/lbm}^\circ\text{F})\ (6\text{hr})\ (200\ \text{ft}^2)} = 12.5^\circ\text{F}$$

Q_L from C-1 for month selected

N_o number of days in month

C_p specific heat of fluid

θ number of hours of usable sun in day, use 5 or 6 hours

G is flowrate in lbm/ft^2hr, use (10 lbm/hr ft^2)

f from D-2 for month selected.

WORKSHEET E-2

PRESENT WORTH ANALYSIS

JOB NO. ______________

Collector Area ________ ft²

		A	B	C
Year____ (Specify CY or FY)	Year of Analysis	Dollar Value of Fuel Saved in Zeroth Year (Work-sheet E-1)**	Fuel Inflation Factor at ____% Discount at 10% From Table 3-5a	Annual Present Worth (A x B)
	Zeroth*			
	1st			
	2nd			
	3rd			
	4th			
	5th			
	6th			
	7th			
	8th			
	9th			
	10th			
	11th			
	12th			
	13th			
	14th			
	15th			
	16th			
	17th			
	18th			
	19th			
	20th			
	21st			
	22nd			
	23rd			
	24th			
	25th			

Present worth of fuel saved by system (transfer to Worksheet A, Column Y) Σ ______________

* Year for which fuel costs are available if year of construction. Otherwise, escalate fuel costs to year of construction.

** Copy zeroth year fuel saved into each space in Column A.

WORKSHEET F

SOLAR SYSTEM COST ANALYSIS

JOB NO. Example 1

	Item Description	Area 200 ft^2 Space Heating DHW	300 ft^2 DHW Only
(1)	Collector (From Table 2-6, Item 22)	13.65 ft^2	13.65 ft^2
(2)	Storage tank, installed, 360 gal & 540 gal $ 3.50 /gal x 1.8 gal/ft^2 collector	6.30 ft^2	6.30 ft^2
(3)	Auxiliary heating unit, installed cost/ft^2 collector, net of conventional unit cost	2.25 ft^2	2.25 ft^2
(3)	Pumps, pipe, heat exchangers, controls cost/ft^2 collector	4.25 ft^2	4.25 ft^2
	Other (from Worksheet G)	0 ft^2	0 ft^2
	Subtotal	26.45 ft^2	26.45 ft^2
(1)	Collector installation	13.65 ft^2	13.65 ft^2
(4)	TOTAL	$40.10/$ft^2$	40.10/ft^2

(4)

Area	Cost/ft^2	Cost
200	40.10/ft^2	$8,020
300	40.10/ft^2	$12,030

(1) Manufacturer's data, or from Table 2-6 or Table 3-7, plus 80 to 100% for installation

(2) See Table 2-8 and Worksheet D-2, plus $.10 (antifreeze) and $1.50/$ft^2$ (heat exchanger) if applicable.

(3) See Table 3-6.

(4) Transfer totals to Worksheet A.

5.0 DIRECTORY OF SOLAR EQUIPMENT MANUFACTURERS

5.1 Manufacturers Directories

SOLAR ENERGY SOURCEBOOK ... C. W. Martz (ed); Solar Energy Institute of America, Box 9352, Washington, DC 20005, 1977, 712 pp, $12.00 (free to members).

Organized compilation of solar energy related products and services in the form of a loose leaf binder; continual updates provided to members.

SOLAR INDUSTRY INDEX ... Solar Energy Industries Association, 1001 Connecticut Ave., N.W., Washington, DC 20036, 1977, 381 pp, $8.00.

Comprehensive guide to manufacturers and service organizations; updated annually.

SOLAR PRODUCTS SPECIFICATION GUIDE ... Solar Age Magazine, Church Hill, Harrisville, NH 03450

A comprehensive list of manufacturers and services for solar components. Published annually and updated six times per year by subscription.

5.2 Manufacturers of Flat Plate Collectors and Other Solar Hot Water Equipment

The following lists are provided for information only. The Civil Engineering Laboratory does not endorse or recommend the quality or capability of any individual company. There probably are other companies in any particular area. Local telephone directories can be checked. The National Solar Heating and Cooling Information Center often can provide names of companies. Their number is (800) 523-2929, toll free.

MANUFACTURERS OF SOLAR HOT WATER EQUIPMENT

MANUFACTURERS (Nationwide, listed by state)

The following lists are provided for information only. The National Solar Heating and Cooling Information Center, The Franklin Institute Research Laboratories, The U. S. Energy Research and Development Administration and The U. S. Department of Housing and Urban Development do not endorse, recommend or attest to the quality or capability of any products or services or companies and individuals.

These lists are based on information available to the National Center at the time of publication. For an up-to-date list, write to the Center.

ALABAMA

Sun Century Systems
P.O. Box 2036
Florence, AL 35630
205—754-0795

ARIZONA

Arizona Solar Enterprises
6719 E. Holly St.
Scottsdale, AZ 85257
602—945-7477

Hansberger Refrigeration & Electric Co.
2450 8th St.
Yuma, AZ 85364
602—783-3331

Helio Associates
P.O. Box 17960
Tucson, AZ 85731
602—792-2800

Mel Kiser and Associates
6701 E. Kenyon Dr.
Tucson, AZ 85710
602—747-1552

Sunpower Systems Corp.
2123 South Priest Rd./Suite 216
Tempe, AZ 85282
602—968-7425

CALIFORNIA

Alten Associates Inc.
2594 Leghorn St.
Mountain View, CA 94043
415—969-6474

American Sun Industries
3477 Old Conejo Rd.
P.O. Box 263
Newbury Pk., CA 91320
805—498-9700

Applied Sol Tech Inc.
P.O. Box 9111 Cabrillo Station
Long Beach, CA 90810
213—426-0127

Baker Bros. Solar Collectors
207 Cortez Ave.
Davis, CA 95616
916—756-4558

Conserdyne Corp.
4437 San Fernando Rd.
Glendale, CA 91204
213—246-8409

Elcam, Inc.
5330 Debbie La.
Santa Barbara, CA 93111
805—964-8676

Energy Systems Inc.
634 Crest Dr.
El Cajon, CA 92021
714—447-1000

Fred Rice Productions
48780 Eisenhower Dr.
P.O. Box 643
La Quinta, CA 92253
714—564-4823

Grundfos
2555 Clovis Ave.
Clovis, CA 93612
209—299-9741

Helio-Dynamics Inc.
518 S. Van Ness Ave.
Los Angeles, CA 90020
213—384-9853

Heliotrope General
3733 Kenora Dr.
Spring Valley, CA 92077
714—460-3930

Kessel Insolar System
2135 Mono Way
Sonora, CA 95370
209—532-2996

Natural Energy Systems
1632 Pioneer Way
El Cajon, CA 92020
714—440-6411

Piper Hydro Corp.
2895 East La Palma
Anaheim, CA 92806
714—630-4040

Powell Brothers Inc.
5903 Firestone Blvd.
South Gate, CA 90280
213—869-3307

Ra-Los Inc.
559 Union Ave.
Campbell, CA 95008
408—371-1734

Raypak Inc.
31111 Agoura Rd.
Westlake Village, CA 91359
213—889-1500

Rho Sigma
15150 Raymer St.
Van Nuys, CA 91405
213—342-4376

Skytherm Processes Engrg.
2424 Wilshire Blvd.
Los Angeles, CA 90057
213—389-2300

Sol-Aire
46 Las Cascadas
Orinda, CA 94563
415—254-2672

Solar-Aire
82 S. Third St.
San Jose, CA 95113
408—295-2528

Solar Applications, Inc.
7926 Convoy Ct.
San Diego, CA 92111
714—292-1857

Solarcoa
2115 E. Spring St.
Long Beach, CA 90808
213—426-7655

Solar Energy Digest
P O Box 17776
San Diego CA 92117
714 - 277-2980

Solar Energy Systems Inc.
336 East Carson St
Carson CA 90049
213—549-4012

Solar II Enterprises
21416 Bear Creek Rd
Los Gatos CA 95030
408 354-3353

Solargenics Inc.
9713 [illegible] Ave
Chatsworth CA 91311
213 [illegible]98-0806

Solar Master Solar Panels
[illegible] D W Betteravia Rd
Santa Maria CA 93454
[illegible]

Solar Utilities Co.
[illegible] N Cedros
Solana Beach CA 92075

Solarway
P O Box 217
Redwood Valley CA 95470
707 485 7616

Solergy Inc
[illegible] St
San Francisco CA 94107
415 495-4303

Sunburst Solar Energy Inc.
P O Box 2799
Menlo Park CA 94025
415 -327-8022

The Sundu Co
3319 Keys La
Anaheim CA 92804
714 828-2873

Sunshine Utility Co.
1444 Pioneer Way Suites 9 & 10
El Cajon CA 92020
714— 440-3151

Sunsource Inc.
1291 S Brass Lantern Dr
La Habra CA 90631
213 -943-8883

Sunwater Energy Products
1488 Pioneer Way = 17
El Cajon CA 92020
714—442-1532

Unit Span Architectural Systems
9419 Mason Ave
Chatsworth CA 91311
213- 998-1131

Western Energy Inc.
454 Forest Ave
Palo Alto CA 94302
415 - 327-3371

Ying Manufacturing Corp.
1957 West 144th St
Gardena CA 90249
213 327-8399

COLORADO

Design Works
P O Box 70[illegible]
Telluride CO 81435

Energy Dynamics Corp.
327 West Vermijo Rd
Colorado Springs CO 80903
303 475-0332

Enertech Corp.
R D 4 P O Box 409
Golden CO 80401
303—642-3891

Entropy Limited
P O Box 2206
Boulder CO 80306
303—443-3319

Future Systems Inc.
12500 W Cedar Dr
Lakewood CO 80228
303 -989-0431

Miromit/American Heliothermal Corp.
3515 S Tamarac Dr Suite 360
Denver CO 80237
303- 773-6085

R. M. Products
5010 Cook St
Denver CO 80216
303—825-0203

Solar Energy Research Corp.
1228 15th St
Denver CO 80202
303—573-5499

Solaron Corp.
300 Gallerina Tower
720 S Colorado Blvd
Denver, CO 80222
303—759-0101

CONNECTICUT

Falbel Energy Systems Corp.
P O Box 6
Greenwich CT 06830
203—357-0626

Hubbell
45 Seymour St
Stratford, CT 06497
203—378-2659

International Environmental Energy Inc.
275 Windsor St
Hartford, CT 06120
203—249-5011

Solar Heating Systems Corp.
151 John Downey Dr
New Britain, CT 06051
203—224-2164

Solar Industries Inc.
100 Captain Neville Dr
Waterbury, CT 06705
203—753-1195

Spiral Tubing Corp.
544 John Downey Dr
New Britain, CT 06051
203—244-2409

Sunworks
P O Box 1004
New Haven CT 06508
203—934-6301

Wilson Solar Kinetics Corp.
P O Box 17308
West Hartford, CT 06117
203—233-4461

DELAWARE

DuPont Co.
Nemours Bldg /Rm 24751
Wilmington, DE 19898
302—999-3456

Solar Systems Inc.
323 Country Club Dr
Rehoboth Beach, DE 19971
302—227-2323

MANUFACTURERS

DISTRICT OF COLUMBIA

Business and Technology, Inc.
2800 Upton St. N W
Washington, DC 20008
202—362-5591 or
202—244-4902

McCombs Solar Co.
1629 K St., N W Suite 520
Washington, DC 20006
202—296-1540

Natural Energy Corp.
1001 Connecticut Ave N W
Washington, DC 20036
202—296-7070

Solartherm
1640 Kalmia Rd., N.W
Washington, DC 20012
202—882-4000

Thomason Solar Homes Inc.
609 Cedar Rd S E
Washington DC 20022
301—336-4042

FLORIDA

American Solar Power, Inc.
715 Swann Ave.
Tampa, FL 33606
813—251-6946

Astro Solar Corp.
744 Barnett Dr./Unit No. 6
Lake Worth, FL 33461
305—965-0606

Aztec Solar Co.
P.O. Box 272
Maitland, FL 32751
305—628-5004

Beutels Solar Heating Co.
7161 Northwest 74th St
Miami, FL 33166
305—885-0122

D. W. Browning Contracting Co.
475 Carswell Ave
Holly Hill, FL 32017
904—252-1528

Capital Solar Heating Inc.
376 N.W. 25th St
Miami, FL 33127
305—576-2380

Chemical Processors Inc.
PO Box 10636
St Petersburg, FL 33733
813—822-3689

Consumer Energy Corp.
4234 S W 75th Ave
Miami FL 33155
305—266-0124

CSI Solar Systems Div.
12400 49th St.
Clearwater, FL 33520
813—577-4228

D & J Sheet Metal Corp.
10055 N W. 7th Ave
Miami, FL 33150
305—757-7033

Del Sol Control Corp.
11914 U S Highway No 1
Juno FL 33408
305—626-6116

Energy Applications Inc
840 Margie Dr.
Titusville, FL 32780
305—269-4893

Falkner Inc.
6121 Alden Rd./PO Box 673
Orlando, FL
305—898-2541

Flagala Corp.
9700 W. Highway 98
Panama City, FL 32401
904—234-6559

Florida Solar Power Inc.
P.O. Box 5846
Tallahassee, FL 32301
904—244-8270

General Energy Devices
1743 Ensley Ave
Clearwater, FL 33516
813—586-1146

Gulf Thermal Corp.
PO Box 13124 Airgate Branch
Sarasota, FL 33580
813—355-9783

Hill Bros. Inc./Thermal Div.
3501 N W 60th St
Miami, FL 33142
305—693-5800

J & R Simmons Construction Co.
2185 Sherwood Dr
South Daytona, FL 32019
904—677-5832

Largo Solar Systems Inc.
2525 Key Largo La
Mail Only
Fort Lauderdale FL 33312
305—583-8090

Matthews Systems Inc
PO Box 1666
Gainesville, FL 32602
904—376-5222

National Solar Systems
PO Box 17348
Tampa FL 33682
813—933-4382

OEM Products Inc./
Solarmatic Div.
2413 Garden St
Tampa, FL 33605
813—247-5858

Semco
1091 S W 1st Way
Deerfield Beach, FL 33441
305—427-0040

Solar Comfort Inc.
249 S Grove St
Venice, FL 33595
813—485-0544

Solar Controlar Inc.
PO Box 8703
Orlando, FL 32806

Solar Development Inc.
4180 West Roads Dr
West Palm Beach FL 33407
305—842-8935

Solar Dynamics Inc.
PO Box 3457
Hialeah, FL 33013
305—921-7911

Solar Electric International
6123 Anno Ave
Orlando, FL 32809

Solar Energy Components Inc.
1605 North Cocoa Blvd
Cocoa, FL 32922
305—632-2880

MANUFACTURERS

Solar Energy Products Inc.
1208 N W 8th Ave
Gainesville, FL 32601
904-377-6527

Solar Energy Resources Corp.
10639 S W 185 Terr
Miami, FL 33157
305-233-0711

Solar Energy Systems
1243 South Florida Ave
Rockledge, FL 32955
305-632-6251

Solar-Eye Products, Inc.
1300 N W McNabe Rd
Bldg GNH
Ft Lauderdale FL 33307
305-974-2500

Solar Fin Systems
140 S Dixie Hwy
St Augustine FL
904-824-3522

Solar Heating & Air Conditioning Systems
13584 49th St North
Clearwater, FL 33520
813-577-3961

Solar Industries of Florida
P O Box 9013
Jacksonville, FL 32208
904-768-4323

Solar Innovations
412 Longfellow Blvd
Lakeland, FL 33801
813-688-8373

Solar Products Inc./Sun-Tank
614 N W 62nd St
Miami, FL 33150
305-756-7609

Solar Systems by Sundance Corp.
4815 S W 75th Ave
Miami, FL 33101
305-264-1894

Solar Water Heaters of New Port Richey
540 Palmetto
New Port Richey, FL 33552
813-848-2343

Southern Lighting/ Universal 100 Products
501 Elwell Ave
Orlando, FL 32803
305-894-8851

Sun Power
10400 S.W. 187th St
Miami, FL 33157
305-233-2224

Sunseeker Systems Inc.
100 W Kennedy Blvd
Tampa, FL 33602
813-223-1787

Systems Technology Inc.
P.O. Box 337
Shalimar, FL 32579
904-863-9213

Unit Electric Control Inc./ Sol-Ray Div.
130 Atlantic Dr
Maitland, FL 32751
305-831-1900

Universal Solar Energy Co.
1802 Madrid Ave.
Lake Worth, FL 33461
305-586-6020

Wilcon Corp.
3310 S.W. Seventh
Ocala, FL 32670
904-732-2550

Wilcox Manufacturing Corp.
P.O. Box 455
Pinellas Park, FL 33565
813-531-7741

W. R. Robbins & Sons
1401 N.W. 20th St.
Miami, FL 33142
305-325-0880

Youngblood Company, Inc.
1085 N.W. 36th St.
Miami, FL 33127
305-635-2501

GEORGIA

Independent Living Inc.
5715 Buford Hwy., N.E.
Doraville, GA 30340
404-455-0927

Scientific-Atlanta Inc.
3845 Pleasantdale Rd
Atlanta, GA 30340
404-449-2000

Solar Technology Inc.
3927 Oakclif Industrial Ct
Atlanta, GA 30340
404-449-0900

Southeastern Solar Systems
P O Box 44066
Atlanta, GA 30336
404-691-1864

Wallace Company
831 Dorsey St
Gainsville, GA 30501
404-534-5971

HAWAII

The Solaray Corp.
2414 Makiki Heights Dr
Honolulu, HI 96822
808-533-6464

ILLINOIS

Amcon Inc.
211 W. Willow St.
Carbondale, IL 62901
618-457-3022

A.O. Smith Corp.
P.O. Box 28
Kankakee, IL 60901
815-933-8241

Chamberlain Manufacturing Corp.
845 Larch Ave.
Elmhurst, IL 60126
312-279-3600

ITT Fluid Handling Div.
4711 Golf Rd.
Skokie, IL 60076
312-677-4030

Johnson Controls Inc./Penn Div.
2221 Camden Ct.
Oak Brook, Il 60521
312-654-4900

Olin Brass Corp./Roll-Bond Div.
E Alton, IL 62024
618-258-2000

MANUFACTURERS

Pak-Tronics Inc.
4044 N Rockwell Ave
Chicago, IL 60618
312--478-8585

Solar Dynamics Corp.
550 Frontage Rd
Northfield, IL 60093
312- 446-5242

Sun Systems Inc.
PO Box 155
Eureka, IL 61530
309—685-9728

INDIANA

Solar Energytics, Inc.
PO Box 532
Jasper, IN 47546
812—482-1416

IOWA

Lennox Industries Inc.
200 S 12th Ave
Marshalltown, IA 50158
414--754-4011

Pleiad Industries, Inc.
RR 1 PO Box 57
West Branch, IA 52358
319—643-5650

Solar Aire
PO Box 276
North Liberty, IA 52317
319—626-2343

KENTUCKY

Mid-Western Solar Systems
2235 Irvin Cobb Dr
P.O. Box 2384
Paducah, KY 42001
502—443-6295

MARYLAND

KTA Corp.
12300 Washington Ave.
Rockville, MD 20852
201—568-2066

MASSACHUSETTS

Columbia Solar Energy Div.
55 High St.
Holbrook, MA 02343
617—767-0513

Daystar Corp
90 Cambridge St
Burlington, MA 01803
617 -272-8460

Diy-Sol Inc.
PO Box 614
Marlboro, MA 01752

Kennecott Copper Corp
128 Spring St
Lexington, MA 021[illegible]
617 862-8268

Sunkeeper
PO Box 34
Shawsheen Village Sta
Andover, MA 018[illegible]
617 —470-0555

Sunsav Inc.
9 Mill St
Lawrence, MA 0184[illegible]
617--686-8040

Sun Systems Inc.
PO Box 347
Milton, MA 02186
617— 268-8178

Vaughn Corp./Solargy Systems
386 Elm St
Salisbury, MA 01950
617—462-6683

MICHIGAN

Dow Chemical, USA
2020 Dow Center
Midland, MI 48640
517—636-3993

Solarator Inc.
16231 W 14 Mile Rd
Birmingham, MI 48009
313—642-9377

Solar Research
525 N. Fifth St
Brighton, MI 48116
313—227-1151

Solartran Co.
Escanaba, MI 49829
906—786-4550

Tranter
735 East Hazel St
Lansing, MI 48909
517—372-8410

MINNESOTA

A To Z Solar Prooucts
200 E 26th St
Minneapolis, MN 554[illegible]
[illegible]

Hoffman Products, Inc
[illegible]
[illegible]
[illegible]

Honeywell Inc
2600 Ridgeway Pkwy
Minneapolis, MN [illegible]
[illegible]

Ilse Engineering
[illegible]
Duluth, MN [illegible]
218 [illegible]-6858

Sheldahl/Advanced Products Div.
Northfield, MN 550[illegible]
[illegible] 645-5633

Solargizer Corp.
220 Mulberry St
Stillwater, MN 55082
612 439-5734

NEVADA

Richdel Inc.
PO Drawer A
Carson City, NV 897[illegible]
702 - 882-6786

S. W. Ener-Tech Inc.
3030 S Valley View B[illegible]
Las Vegas, NV 89102
702 - 873-1975

NEW HAMPSHIRE

Heliopticon Corp.
PO Drawer 330
Plymouth, NH 03264
603—536-1070

Kalwall Corp./
Solar Components Div.
PO Box 237
Manchester, NH 03105
603—668-8186

MANUFACTURERS

NEW JERSEY

Berry Solar Products
Woodbridge At Main
PO Box 327
Edison NJ 08817
201 549-3800

Calmac Manufacturing Corp.
PO Box 710E
Englewood NJ 07631
201 569-0420

Heilemann Electric
127 Mountain View Rd
Warren NJ 07060
201 -757-4507

Solar Energy Systems Inc.
One Olney Ave
Cherry Hill NJ 08003
609 424-4446

SSP Associates
704 Blue Hill Rd
River Vale NJ 07675
201 - 391-4724

NEW MEXICO

K-Line Corp.
911 Pennsylvania Ave
Albuquerque NM 87110
505 268-3379

Sigma Energy Products
1405B San Mateo N E
Albuquerque NM 87110
505–262-0516

NEW YORK

Advance Cooler Manufacturing Corp.
Route 146
Bradford Industrial Park
Clifton Park, NY 12065
518–371-2140

Ford Products Corp.
Ford Products Rd
Valley Cottage, NY 10989
914–358-8282

Grumman Corp./ Energy Sys. Div.
Dept GR
4175 Veterans Memorial Hwy
Ronkonkoma NY 11779
516–575-7062

Hitachi American Ltd.
437 Madison Ave
New York NY 10022
212 838-4804

International Environment Corp.
129 Halstead Ave
Mamaroneck NY 10543
914–698-8130

Revere Copper & Brass Inc.
PO Box 151
Rome NY 13440
315–338-2401

Solar Energy Systems, Inc.
P O Box 625 Sentry Pl
Scarsdale, NY 10583
914–725-5570

Sol-Therm Corp.
7 West 14th St
New York, NY 10011
212–691-4632

NORTH CAROLINA

Carolina Solar Equipment Co.
PO Box 2068
Salisbury, NC 28144
704–637-1243

Standard Electric Co.
PO Box 631
Rocky Mount NC 27801
919–442-1155

OHIO

Glass-Lined Water Heater Co.
13000 Athens Ave
Cleveland, OH 44107
216–521-1377

Howard Bell Enterprises Inc.
PO Box 413
Valley City, OH 44280
216–483-3249

Libbey Owens Ford/ Technical Center
1701 East Broadway
Toledo, OH 43605
419–247-4355

Mor-Flo Industries Inc.
18450 South Miles Rd
Cleveland, OH 44128
216–663-7300

Owens Illinois Inc.
PO Box 1035
Toledo OH 43666
419–242-6543

Ranco Inc.
601 W Fifth Ave
Columbus OH 43201
614–294-3511

Solar Energy Products Co.
121 Miller Rd
Avon Lake, OH 44012
216–933-5000

Solar Heat Corp.
1252 French Ave
Lakewood OH 44107
216–228-2993

Solar Home Systems Inc.
12931 West Geauga Trail
Chesterland, OH 44026
216–729-9350

Solar Vak Inc.
PO Box 1444
Dayton, OH 45414
513–278-6551

OKLAHOMA

Brown Manufacturing Co.
PO Box 14546
Oklahoma City, OK 73114
405–751-1343

Tri-State Solar King, Inc.
PO Box 503
Adams, OK 73901
405–253-6562

OREGON

Scientifico Components Co.
35985 Row River Rd
Cottage Grove, OR 97424

PENNSYLVANIA

Aluminum Co. of America
Alcoa Bldg
Pittsburgh, PA 15219
412–553-2321

Ametek Inc./ Power Systems Group
1 Spring Ave
Hatfield, PA 19440
215–822-2971

MANUFACTURERS

Enviropane Inc.
348 N Marshall St
Lancaster, PA 17602
717—299-3737

General Electric Co.
PO Box 8661, Bldg 7
Philadelphia, PA 19101
215—962-2112

Heliotherm Inc.
Lenni, PA 19052
215—459-9030

Packless Industries Inc.
PO Box 310
Mount Wolf, PA 17347
717—266-5673

PPG Industries
One Gateway Center
Pittsburgh, PA 15222
412—434-3555

Practical Solar Heating
209 S. Delaware Dr /Rt 611
Easton, PA 18042
215—252-6381

Simons Solar Environmental Systems Inc.
24 Carlisle Pike
Mechanicsburg, PA 17055
717—697-2778

Solar Heat Co.
PO Box 110
Greenville, PA 16125
412—588-5650

Solar Shelter
P.O. Box 36
Reading, PA 19603
215—488-7624

Sun Earth Solar Products Corp.
RD 1/PO Box 337
Green Lane, PA 18054
215—699-7892

Sunwall Inc.
P.O. Box 9723
Pittsburgh, PA 15229
412—364-5349

RHODE ISLAND

Solar Homes Inc.
2 Narragansett Ave
Jamestown, RI 02835
401—423-1025

TENNESSEE

ASG Industries
P.O. Box 929
Kingsport, TN 37662
615—245-0211

Energy Converters Inc.
2501 N Orchard Knob Ave
Chattanooga, TN 37406
615—624-2608

Oak Ridge Solar Engineering Inc.
PO Box 3016
Oak Ridge, TN 37830
615—482-5290

State Industries Inc.
Cumberland St
Ashland City, TN 37015
615—792-4371

W. L. Jackson Manufacturing Co.
PO Box 11168
Chattanooga, TN 37401
615—867-4700

TEXAS

American Solar King Corp.
6801 New McGregor Hwy
Waco, TX 76710
817—776-3860

Cole Solar Systems Inc.
440A E St Elmo Rd
Austin, TX 78745
512—444-2565

Northrup Inc.
302 Nichols Dr
Hutchins, TX 75141
214—225-4291

Solar Systems Inc.
507 W Elm St
Tyler, TX 757?1
214—592-5343

Soltex Corp.
PO Box 55703
Houston, TX 77055
713—780-1733

Solus, Inc.
PO Box 35227
Houston, TX 77035
713—772-6416

VERMONT

Sol-R-Tech
PO Box G
Hartford, VT 05047
802—295-9342

VIRGINIA

Atlantic Solar Products Inc.
Reston International Center
Suite 227
11800 Sunrise Valley Dr
Reston, VA 22091
703—620-2300

Helios Corp.
1313 Belleview Ave
Charlottesville, VA 22901
804—293-9574

Reynolds Metals Co.
PO Box 27003
Richmond, VA 23261
804—281-3026

Solar American
106 Sherwood Dr
Williamsburg, VA
804—229-0657

Solar Corp. of America/ Intertechnology Corp.
100 Main St
Warrenton, VA 22186
703—347-7900

Solar Energy Co.
PO Box 649
Gloucester Point, VA 23062

Solar One Ltd.
709 Birdneck Rd
Virginia Beach, VA 23451
804—422-3262

Solar Sensor System
4220 Berritt St
Fairfax, VA 22030
703—273-2683

WASHINGTON

Ecotope Group
747 16th Street, E
Seattle, WA 98112
206—322-3753

E&K Service Co.
16824 74th Ave., N E
Bothell, WA 98011
206—486-6660

WISCONSIN

Solaray Inc.
324 S Kidd St
Whitewater, WI 53190
414—473-2525

Sun Stone
PO Box 941
Sheboygan, WI 53081
414—452-8194

6.0 REFERENCES (●) AND SELECT BIBLIOGRAPHY

AIA Research Corp. (1979). A survey of passive solar buildings. San Mateo, Calif., Solar Energy Information Services, 1979.

American Society of Heating, Refrigerating, and Air-Conditioning Engineers (1979). ASHRAE handbook of fundamentals, chapter 22. New York, N.Y., American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 1979.

● Anderson, B. (1976). Solar home book. Harrisville, N.H., Cheshire Books, 1976.

Anderson, L. B., Hottel, H. C., and Whillier, A. (1955). Solar energy research no. 47, Madison, Wis., University of Wisconsin, 1955.

● Anderson, M. (1979). "Accolade at Honeywell Plaza," Solar Age, vol 4, no. 6, Jun 1979, pp 32-36.

Andrews, J. W. and Wilhelm, W. (1977). Low cost site assembled solar collector designs for use with heat pumps. Brookhaven National Laboratory, Report no. BNL 50671, Upton, N.Y., May 1977.

Antolini, H. L., et al. (1978). Sunset homeowner's guide to solar heating. Menlo Park, Calif., Lane Publishing Co., 1978.

● Bainbridge, D. A. (1978). "Natural cooling in California," in Proceedings, 1978 Annual Meeting of the American Section, International Solar Energy Society, Denver, Colo., Aug 28-31, 1978. Newark, Del., International Solar Energy Society, 1978, vol 2.1, pp 475-487.

Balcomb, D., et al. (1975). Solar heating handbook for Los Alamos. Los Alamos Scientific Laboratory, Report no. LA-5967, Los Alamos, N.M., May 1975.

- Balcomb, J. D. (undated). "Rules of thumb for passive solar heating in New Mexico," New Mexico Solar Energy Association Bulletin, vol 2, nos. 5, 6, and 7, by J. Douglas Balcomb of Los Alamos Scientific Laboratories.

Balcomb, J. D., Hedstrom, J. C., and Rogers, B. T. (1975). "Design considerations of air cooled collector/rock-bin storage solar heating systems," in International Solar Energy Conference and Exposition, Los Angeles, Calif., Jul 28-Aug 1, 1975. Rockville, Md., International Solar Energy Society, 1975, pp 314-315.

- Barber, R. E. (1975). "Design and test of a prototype 3-ton solar heated Rankine cycle air conditioner," in 1975 International Solar Energy Congress and Exposition, Los Angeles, Calif., Jul 28-Aug 1, 1975. Rockville, Md., International Solar Energy Society, 1975, pp 405-406.

- Barnaby, C. W., et al. (1977). Solar for your present home, prepared for the California Energy Commission, Berkeley, Calif., Berkeley Solar Group, 1977.

Beard, J. T. et al. (1976). "Performance and analysis of 'Solaris' water-trickle solar collector," in Sharing the Sun: Solar Technology in the Seventies, Joint Conference of the International Solar Energy Society, American Section and Solar Energy Society of Canada, Inc., Winnipeg, Manitoba, Aug 15-20, 1976. Cape Canaveral, Fla., International Solar Energy Society, American Section, 1976, vol 2, pp 46-63.

- Beck, E. J. and Field, R. L. (1977). Solar heating of buildings and domestic hot water. Civil Engineering Laboratory, Report R-835, Port Hueneme, Calif., Nov 1977. (ADA054601)

- Beckman, W. A. (1977). "Problems of solar cooling systems," Sunworld, no. 6, Nov 1977, pp 2-6.

• Beckman, W. A., Klein, S. A., and Duffie, J. A. (1977). Solar heating design by the F-chart method, New York, Wiley-Interscience, 1977.

Braden, S. (1977). Graphic standards of solar energy. Boston, Mass., CBI Publishing Co., 1977.

Cole-Appel, B. E. and Haberstroh, R. D. (1976). "Performance of air-cooled flat plate collectors," in Sharing the Sun: Solar Technology in the Seventies, vol 3, pp 116-117.

• Cole, R. L., et al. (1979). Design and installation manual for thermal energy storage. Argonne National Laboratory, Report no. ANL-79-15, Argonne, Ill., Feb 1979.

Consroe, T., et al. (1974). Alternate strategies for optimizing energy supply, distribution, and consumption systems on Naval bases, vol 2. Advanced energy conservation strategies. Civil Engineering Laboratory, Contract Report CR 74.007, Port Hueneme, Calif., Jan 1974. (AD786757, vol 1 published as CEL CR 74.006, Nov 1973, AD777471).

De Winter, F., (1973). Fin effectiveness or efficiency in flat plate solar collectors. New York, N.Y., Copper Development Association, 1973.

• Duffie, J. A. and Beckman, W. A. (1974). Solar energy thermal processes. New York, Wiley-Interscience, 1974.

Duffie, J. A., Beckman, W. A., and Dekken, J. G. (1977). "Solar heating in North America," Mechanical Engineering, vol 99, no. 11, Nov 1977.

Duffie, J. A., Chung, R., and Lof, G. O. G. (1963). "A study of a solar air conditioner," Mechanical Engineering, vol 85, no. 8, Aug 1963, pp 31-35.

● Durlak, E. R. (1979a). Analysis of solar collectors, type: double glazed glass cover with aluminum absorber plate, copper waterways, flat black coating. Civil Engineering Laboratory, Technical Memorandum M63-79-16, Port Hueneme, Calif., Jul 1979.

● Durlak, E. R. (1979b). Computer programs that simulate solar system performance. Civil Engineering Laboratory, Techdata Sheet 79-06, Port Hueneme, Calif., Apr 1979.

Edwards, D. K. (1977). Solar collector design. Philadelphia, Pa., Franklin Institute Press, 1977.

Edwards, D. K., et al. (1960). Basic studies on the use and control of solar energy. Los Angeles, Calif., University of California. Engineering Report no. UCLA 60-93, Oct 1960.

Energy Research and Development Administration (1975). Catalog on solar energy heating and cooling products. Washington, D.C., Oct 1975.

Energy Research and Development Administration (1976). An economic analysis of solar water and space heating. DSE-2322-1, Washington, D.C., Nov 1976.

Energy Research and Development Administration (1977). ERDA facilities solar design handbook. Prepared by Los Alamos Scientific Laboratory, Group WX-4, Washington, D.C., Aug 1977.

Energy Research and Development Administration, San Francisco Operations Office (1976). Pacific regional solar heating handbook, 2nd ed. Washington, D.C., GPO, Nov 1976.

● Eyre, P. E. (1978). Corrosion in solar systems. Report done on CEL contract nos. N62583/77 MR 861 and N62583/77 MR 881. Port Hueneme, Calif., Dec 1978.

Federal Energy Administration (1976). Buying solar. Report no. FEA/G-76/154, Washington, D.C., Jun 1976. (PB 262134)

Florida Solar Energy Center (1979). Summary test results FSEC certified solar collectors. Cape Canaveral, Fla., Feb 1979.

Foster, W. M. (1977). "Design options for residential solar hot water heating systems," Solar Engineering, Oct 1977, pp 30-32.

Garg, H. P. (1974). "Effect of dirt on transparent covers in flat plate solar energy collectors," Solar Energy, vol 15, no. 4, Apr 1974, pp 299-302.

Gupta, G. L. and Garg, H. P. (1968). "System design in solar water heaters with natural circulation," Solar Energy, vol 12, 1968, pp 163-182.

Hill, J. E. and Kusada, T. (1974). Method of testing for rating solar collectors based on thermal performance. National Bureau of Standards, Report no. NBSIR 74-635, Washington, D.C., 1974.

Hottel, H. C., and Woertz, B. B. (1942). "The performance of flat-plate collectors," American Society of Mechanical Engineers, Transactions, vol 64, no. 2, Solar-Heat, Feb 1942, pp 91-104.

Joncich, D., et al. (1978). Design of solar heating and cooling systems. Champaign, Ill., Construction Engineering Research Laboratory, Technical Report E-139. Oct 1978.

Kimbell, B. W. (1978). Recommendation of preferred solar heating system combinations. Report done on CEL contract nos. N62583/77 MR 861 and N62583/77 MR 881. Port Hueneme, Calif., Dec 1978.

Kirts, R. E. (1978). Technical and economic evaluation of four solar assisted heat pump systems. Civil Engineering Laboratory, Technical Memorandum M63-78-6, Port Hueneme, Calif., Aug 1978.

- Klein, S. A., Beckman, W. A., and Duffie, J. A. (1976). "A design procedure for solar heating systems," Solar Energy, vol 18, no. 2, 1976, pp 113-127.

- Klein, S. A., Beckman, W. A., and Duffie, J. A. (1977). "A design procedure for solar air heating systems," Solar Energy, vol 19, no. 5, 1977, pp 509-512.

- Kohler, J. (1978). "Rules of thumb for sizing storage," Solar Age, vol 3, no. 4, 1978.

Kreider, J. J. and Kreith, F. (1975). Solar heating and cooling: engineering, practical design, and economics. New York, McGraw-Hill, 1975.

Liu, B. H. Y. and Jordan, R. C. (1963). "A rational procedure for predicting the long-term average performance of flat-plate solar energy collectors," Solar Energy, vol 7, no. 2, Apr-Jun 1963, pp 53-74.

- Lof, G. O. G. (1977). "Systems for space heating with solar energy," in Applications of Solar Energy for Heating and Cooling of Buildings, ed. by R. C. Jordan and B. Y. H. Liu. New York, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 1977, XII, 19 pp.

Lof, G. O. G. and Nevens, T. D. (1953). "Heating of air by solar energy," Ohio Journal of Science, vol 53, 1953, pp 272-280.

Lof, G. O. G. and Tybout, R. A. (1973). "Cost of house heating with solar energy," Solar Energy, vol 14, no. 3, Feb 1973, pp 253-255.

Lof, G. O. G., Karaki, S., and Smith, C. C. (1976). "Comparative performance of solar heating with air and liquid systems," in Sharing the Sun: Solar Technology in the Seventies, vol 3, pp 106-117.

- Mazria, E. (1979). Passive solar energy book. Emmaus, Pa., 1979.

- McCullagh, J. C. (1978). Solar greenhouse book. Emmaus, Pa., 1978.

McGarity, A. E. (1977). Solar heating and cooling: an economic assessment. Washington, D.C., National Science Foundation, 1977.

- Merriam, R. (1977). Solar air conditioning study. Civil Engineering Laboratory, Contract Report CR 77.018, Port Hueneme, Calif., Apr 1977. (ADA043951)

- Montgomery, R. H. (1978). Solar decision book: your guide to making a sound investment. Midland, Mich., Dow Corning Corp., 1978.

National Aeronautics and Space Administration (1974). Harnessing the sun's energy for heating and cooling. George C. Marshall Space Flight Center, Ala., 1974.

- National Solar Heating and Cooling Information Center (1979). Solar hot water and your home, Rockville, Md., Aug 1979. (HUD-PDR-466)

Obrecht, M. F. and Myers, J. R. (1973). "Performance and selection of materials for potable hot water service," Heating, Piping, and Air Conditioning, vol 45, Aug 1973. (Also in: Materials Protection and Performance, vol 12, no. 4, Apr 1973, pp 17-24)

Office of Technology Assessment (1978). Application of solar technology to today's energy needs. Washington, D.C., Jun 1978. (2 vols)

Pesko, C., ed. (1975). Solar directory. Ann Arbor, Mich., Ann Arbor Science Publishers, 1975.

Pickering, E. E. (1975). "Residential hot water solar energy storage," in Proceedings of the Workshop on Solar Energy Storage Subsystems for Heating and Cooling of Buildings, Charlottesville, Va., Apr 16-18, 1975, ed. by L. U. Lilleleht. Washington, D.C., National Science Foundation, 1975.

Rho-Sigma (undated). [Brochure], 11922 Valerio Street, North Hollywood, Calif.

Scholten, W. and Curran, H. (1979). "Active cooling update," Solar Age, vol 4, no. 6, Jun 1979, pp 37-41.

Selcuk, K. (1971). "Thermal and economic analysis of the overlapped glass plate solar air heater," Solar Energy, vol 13, 1971, pp 165-191.

Sharp, M. K. and Loehrke, R. I. (1978). Stratified thermal storage in residential solar energy applications. Report done on CEL contract no. N68305-76-C-0036 and ERDA contract no. EG-72-5-024523.A000. Port Hueneme, Calif., Jun 1978.

Solar Age Magazine (1979). Solar products specification guide. Harrisville, N.H., Solar Vision, Inc., Jul 1979.

Solar Engineering (1978). "The bigger the better for solar in assisting heat pump systems," Solar Engineering, Feb 1978, pp 36-43.

Solar Engineering (1978). "Solar system chemicals," Solar Engineering, Aug 1978, p 42.

Swindler, A. (1979). Size and cost guidelines for solar heating and cooling systems. Report done on CEL contract nos. N62583/77 MR 861 and N62583/77 MR 881. Port Hueneme, Calif., Apr 1979.

Thomason, H. E. and Thomason, H. J. L. (1973). "Solar houses/heating and cooling progress report," Solar Energy, vol 15, no. 1, May 1973, pp 27-39.

Tybout, R. A. and Lof, G. O. G. (1970). "Solar house heating," National Resources Journal, vol 10, 1970, pp 268-326.

U.S. Department of Commerce, Economic Development Administration (1977a). Solar heating and cooling of residential buildings: sizing, installation, and operation of systems. Prepared by the Solar Energy Applications Laboratory, Colorado State University, Washington, D.C., Oct 1977.

U.S. Department of Commerce, Economic Development Administration (1977b). Solar heating and cooling of residential buildings: design of systems. Prepared by the Solar Energy Applications Laboratory, Colorado State University, Washington, D.C., Oct 1977.

U.S. Department of Energy (1978a). Solar heating and cooling of buildings commercialization report, pt. A. Options and strategies. Washington, D.C., May 1978, 3 vols. (Contract CR-05-70065-0)

U.S. Department of Energy (1978b). Report no. NCP/M2528-01: Legal barriers to solar heating and cooling of buildings. Washington, D.C., Mar 1978.

- U.S. Department of Housing and Urban Development (1977). Report no. 4930.2: Solar heating and domestic hot water systems, HUD intermediate minimum property standards, Washington, D.C., 1977.

- U.S. National Oceanic and Atmospheric Administration, National Climatic Center (1968). Climatic atlas of the United States. Asheville, N.C., Jun 1968.

- U.S. Naval Facilities Engineering Command (1978). NAVFAC P-89: Engineering weather data. Washington, D.C., Jul 1978.

Veltford, T. E. (1942). Solar water heaters. New York, N.Y., Copper and Brass Research Association, Mar 1942.

Werden, R. G. and Spielvogel, L. G. (1969). "Sizing of service water heating equipment in commercial and institutional buildings, pt. 2. Research program data acquisition, reduction and analysis, and recommendations for sizing." ASHRAE Transactions, vol 75, 1969, pp 181-199.

Whillier, A. (1963). "Plastic covers for solar collectors," Solar Energy, vol 7, no. 3, Jul-Sep 1963, pp 148-151.

Whillier, A. (1967). "Design factors influencing collector performance," in Low Temperature Engineering Applications of Solar Energy, ed., by R. C. Jordan. New York, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 1967, pp 27-40.

Yellot, J. I., et al. (1974). "Solar energy utilization for heating and cooling," in ASHRAE Handbook, vol 3, Applications. New York, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, 1974, chap. 59.

Yellot, J. I., et al. (1975). Solar-oriented architecture. Tempe, Ariz., Arizona State University, College of Agriculture, 1975.

Zarem, A. M. and Erway, D. D., eds. (1963). Introduction to the utilization of solar energy. New York, McGraw-Hill, 1963.

7.0 LIST OF SYMBOLS

A_c	Collector area (ft^2)
C_F	Cost of fuel ($\$/10^6$ Btu)
C_{min}	Lesser of the heat capacity flow rates in the space heating load exchanger
COP	Coefficient of performance
C_p	Specific heat of fluid (Btu/lbm°F)
f	Fraction of building load supplied by solar heating
$\bar{f}$	Average of f over one year
F_I	Function of energy absorbed by collector/building heating load (ft^{-2})
F_L	Function of collector heat losses/building heating load (ft^{-2})
F_R	Collector heat removal factor (¶ 3.2)
F'_R	Collector heat exchanger efficiency factor (¶ 3.2)
G	Flow rate through collector per unit area (lbm/hr ft^2)
I	Solar insolation (Langley/day)
I_T	Average instantaneous solar insolation on collector surface
L	Gross heat loss rate (Btu/ft^2-degree day)
M	Area of building (ft^2)
$\dot{m}$	Mass flow through collector (lbm/hr)

$(\dot{m}\ C_p)_c$	Heat capacity flow rate through collector (Btu/hr°F)
$(\dot{m}\ C_p)_{min}$	The lesser of the two heat capacity flow rates in the collector-tank heat exchanger
N_o	Number of days in month
N_I	Number of days in computation period
P	Degree days
q_c	Solar heat collected per ft^2 of collector per month (Btu/ft^2mo)
Q_d	DHW heating load (Btu/day)
Q_{dt}	Yearly total load (DHW only) (Btu/yr)
Q_L	Total heat load (space + DHW) per month (Btu/mo)
Q_{L_t}	Yearly total heat load (space + DHW) (Btu/yr)
Q_u	Useful heat collected = $A_c q_c$ (Btu/mo)
R	Gross space heat load = L x M x P
S	Slope factor = ratio of direct solar radiation on a tilted surface to that on a horizontal surface
T_a	Average ambient air temperature (°F)
T_i	Collector inlet fluid temperature (°F)
T_o	Collector outlet fluid temperature (°F)
T_{ref}	212°F, a reference temperature

UA	Overall heat loss coefficient of building times building area (Btu/hr°F)
U_L	Collector overall heat loss coefficient (Langleys/°F-day)
V	Net space heat load = R x ηw
W	Weight of DHW to be heated/day (lbm)
α	Absorptance
ε	Emittance
ε_c	Effectiveness of the collector-tank heat exchanger
ε_L	Effectiveness of the space heating load heat exchanger
η	Collector efficiency
$\eta_{collect}$	Average collector efficiency
$\eta_{delivery}$	Delivery efficiency
η_w	Heater (DHW or space) utilization efficiency
θ	Hours of useful sun/day
$(\tau\alpha)$	Product of cover transmittance and plate absorptance accounting for dirt and shading
$(\overline{\tau\alpha})$	Average value of $(\tau\alpha)$ over one day
$(\tau\alpha)_n$	$(\tau\alpha)$ at normal radiation incidence
ϕ	Utilizability (¶ 3.12)

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The purpose of this document is to provide guidance in the design and cost analysis of solar heating systems for buildings and domestic hot water. Among the topics included are the types of solar systems, components, and solar collectors; discussions on storage systems, heat pumps, cooling systems, passive systems, controls, and pumps. Calculation methods are included for determining collector size, storage size, simplified building and

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DHW loads, value of fuel saved, and saving-investment ratios. The calculation procedure is based on parametric curves for "fraction of heating load supplied by solar energy" and several "rules of thumb" for design. A series of 11 worksheets is used to enable the engineer with no prior experience with solar systems to accomplish a complete design and cost analysis. With this information he can prepare bidding and specification documents for the job. Tables of solar insolation at various Navy stations, typical building heat loads, collector prices by type, and storage tank prices are included. An example problem is worked. A directory of manufacturers and bibliography is also included.

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